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THE BRITISH SHIP RESEARCH ASSOCIATION

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ABSTRACTS *from Current Technical Literature*

The following Abstracts purport to be fair summaries of the articles, but the Association does not accept responsibility for statements made in the originals, nor does it necessarily agree with their contents.

The standard form of reference to the source of each Abstract is: Title of Periodical or Publication (abbreviated according to the list on pp. 3–19 of B.S.R.A. Journal for January 1968), volume number (in heavy type), year, and page number, followed by the date of issue where appropriate. The length of the article and other bibliographical details are also indicated.

SHIP RESISTANCE AND FLUID MOTION

26,778 **Measurements of Components of Resistance on a Tanker Model.** STEELE, B. N. *N.P.L. Ship Report No. 106* (Dec. 1967) [27 pp., 20 ref., 1 tab., 13 graphs, 1 diag.]

The increases in size and fullness of modern tankers, and particularly the increase in displacement obtained by an increase in the beam, pose many hydrodynamic problems. The designer has insufficient information at present upon which to predict such factors as flow separation and bilge vortex formation, which give rise to low hull and propeller efficiencies and severe vibration. The Resistance Research Programme at N.P.L. is attempting to remedy this deficiency. This Report describes experiments on a 10-ft paraffin-wax model of a 0·80 C_b raked-bow tanker.

The Report gives details of measurements of local shear stress, local pressure, wave pattern, and total resistance on the model. The shear-stress and pressure measurements have been integrated over the hull surface to give the frictional and pressure components of resistance. Combination of the results of these measurements suggests that the method of measurement of skin friction, and the hypothesis upon which it is based, are sound. The pressure form effect is found to be large compared with that of a streamlined body.

The methods of measurement and their limitations are described in the Report. Shear-stress and friction-distribution measurements were determined by the Preston Tube technique (see Abstract No. 8754, Apr. 1954). Pressure resistance was accurately determined by measuring the sinkage and trim of the model in addition to the pressures. The wave-pattern resistance was determined by the Eggers-cut technique (see also Abstract No. 25,195, Apr. 1967), and the wave profile was measured by spraying the hull with a p.v.c. aerosol spray; the p.v.c. did not run down below the water surface and defined the wave profile clearly.

After discussion of the results, which are presented graphically, the Report concludes that the measured friction and pressure components sum to the measured total resistance. The results suggest that the "law of the wall" holds in non-uniform pressure gradients. The calibration of the Preston tube obtained by Preston is confirmed. The frictional resistance of the model was shown to be close to the Hughes line. The frictional component of resistance on the model amounted to 64% and the pressure component to 36% of the total resistance at service speed.

The wave-pattern resistance accounted for only 3% of the total resistance at service speed, which implies that the pressure form effect is 33% of the total resistance.

26,779 **The Knuckled Bow-Bulb** (in German). *Hansa*, **104** (1967), p. 1879 (Special Issue in Nov.) [2 pp., 4 graphs, 2 diag.]

This patented type of bow bulb, which is characterised by a knuckle which may be sharp or slightly rounded, was developed by the Paul Lindenau shipyard, Kiel, in collaboration with Professor H. Schneekluth. Model tests have shown that a bulb of this form reduces pitch amplitude, improves seakindliness, and considerably lowers resistance. Some reasons for these favourable properties are mentioned.

Model tests on a 5,720/4,300-ton d.w. shelterdecker of 0·67 C_b, run in 1965–66 at the VWS, Berlin, showed that the knuckled bulb gave a speed increase of 0·13 knots, or allowed a power reduction of 5%, at open-shelterdecker draught (as compared with the no-bulb condition). Corresponding figures for the ballast draught were 0·38 knots and 13%; for closed draught they were 0·08 knots and 2·5%. In each case, sea-keeping was improved.

Tests at the Hamburg tank, on a 2,150-ton d.w. tanker (a 1,000-ton gross paragraph ship) of 0·72C_b, showed that, as compared with the usual type of bulb, the knuckled bulb increased speed by almost ¼ knot or allowed a reduction of 125 s.h.p. (8·5%).

The shelterdecker bulb tested had a sharp knuckle. The knuckle on the tanker bulb was slightly rounded; this bulb was placed comparatively high at the bow. The article includes drawings of both these bulbs, together with power/speed curves from the tests. Model tests on a cargo ship, with various forms of knuckled bulb, are continuing.

26,780 **Turbulence Measurement in Water using Hot-Film Sensors.** SMITH, K. V. H., and RIFAI, M. F. *Civ. Engng*, **63** (1968), p. 981 (Sept.) [3½ pp., 5 ref., 2 graphs, 7 diag.]

Hot-wire anemometers are well suited to measurements of turbulence and mean velocity in flowing fluids. When used in water, however, the ordinary hot-wire anemometer suffers much more from surface contamination, fluid conductivity, and variation of fluid temperature than when used in air; and hot wires therefore do not give consistent results in water without stringent precautions. Many of the difficulties can be greatly reduced by using a sensor consisting of a thin platinum film fused on to a glass surface, with the connecting wires for the heating current embedded in the glass.

The film can be formed on the tip of a wedge (whose forward end may be linear or parabolic) or slightly back from the tip of a conical probe. Experience has shown that the parabolic sensor is the most satisfactory. The development of techniques for spattering a thin layer of quartz over the film has eliminated the need for using high-resistivity water, and has resulted in improved stability.

The Authors describe the constant-resistance method of using hot-film sensors, which has been shown by experience to be better than the constant-current method. The basic Wheatstone-bridge circuit is shown, and the equations relating bridge voltage to the various resistances, including that of the sensor, and to the amplifier gain are derived, as are

the equations relating the current through the sensor to the fluid velocity and the sensor temperature or resistance. Compensation for variations in the water temperature is also explained.

For turbulence measurements the time constant of the sensor, on which depends its lag and response to rapid velocity changes, is important. The Authors explain how the time constant can be determined. A typical parabolic sensor was found to have a time constant of 0·42 milliseconds, which means that its response was satisfactory for turbulent velocity fluctuations of frequency up to 400 c.p.s.

The stability of hot-film sensors has been shown by tests to be satisfactory and considerably better than that of hot wires. A Thermo-Systems quartz-coated parabolic hot-film sensor tested in ordinary tap water with no precautions other than temperature control showed a drop in output voltage of only 1·5% after six hours. A conical Thermo-Systems sensor gave a similar result—a drop of 1% after five hours. These small changes can be allowed for by check tests before and after an investigation. With hot wires drops in output voltage of 4% after five hours and, in another instance, 16% after three hours, were experienced.

In the final section of the article, the Authors explain how hot-film sensors are calibrated to establish the relation between output voltage and the fluid velocity, and describe a constant-temperature anemometer (type ISVR 201) developed by the Institute of Sound and Vibration Research at Southampton University.

PROPELLERS AND PROPULSION

26,781 **KaMeWa Controllable-Pitch Propeller Offers More Profitable Sailing [i.e. Operation].** JOSEFSON, G. *Zosen*, 13 (1968), p. 29 (June) [4 pp., 3 tab., 7 graphs, 1 phot.]

The Author discusses the advantages to be gained in operating efficiency and safety by using controllable-pitch (c.p.) instead of fixed-pitch (f.p.) propellers in ships with automated propulsion systems. A table shows how the number of KaMeWa c.p. propellers in service, and the diameter of the largest of them, have grown since 1948, and a graph illustrates the growing popularity of c.p. propellers for new merchant ships. This trend is not yet apparent in Japan.

At present, most machinery specifications envisage the use of fixed-pitch propellers; alterations to suit c.p. propellers involve extra first-costs which normally can be recovered over a short operating period. However, by treating the c.p. propeller as an integral part of the propulsion machinery in the design stage these extra costs can be greatly reduced and possibly eliminated. Tabular comparisons are made of initial costs (in U.S. dollars) for c.p. and f.p. installations in (*a*) a 100,000-ton d.w. tanker with a 23,000-b.h.p. direct-drive Diesel, and (*b*) a 25,000-ton d.w. bulk carrier with two medium-speed Diesels geared to a single shaft (10,500 s.h.p. at 105 r.p.m.); the estimated annual savings for the c.p. installations are also tabulated. In the case of the bulk-carrier c.p. installation the total first cost is the same as that of the f.p. installation ($246,000), but there is an annual saving in ship-operating costs of $32,000. The initial costs of the tanker installations are $264,000 (c.p.), and $188,000 (f.p.), but this difference should be recovered in about two years.

The main factors which contribute to the higher economic efficiency of a c.p. installation are: (1) Accurate pitch adjustment to suit the particular operating conditions at any time ensures higher average ship speeds; this point is illustrated by graphs comparing the effects of additional resistance, due to hull fouling and wind and sea conditions, on c.p. and on f.p. installations. (2) The prime mover can be non-reversible and can run on heavy fuel at constant speed under all conditions (even when manœuvring); this also allows it to drive a ship's service alternator. (3) Automatic load control of the prime mover reduces maintenance and repair costs. (4) Reduction in berthing costs (tug hire, etc.) due to improved manœuvrability. (5) Possible reduction in the engine-room manning scale. Other, less tangible, advantages of the c.p. installation are its superior stopping ability (this is illustrated by curves for 100,000-ton d.w. tankers), the stepless full-range control of ship-speed, and the possibility of countering hull vibration by adjustment of pitch.

Some of the few disadvantages of the c.p. propeller are discussed; they are insignificant when compared with the advantages. The larger hub ratio (about 0·28) causes a speed loss of only about 0·05 knot; this is demonstrated by curves of propulsion factors against hub ratio.

Propeller maintenance costs are higher. A survey of repairs over a four-year period to the end of 1966, during which a total of 936 propellers were in service, shows that 175 new blades were supplied and 317 blades repaired. This indicates that every propeller requires complete re-blading in the course of 20 years' service, and that the damage rate is at present one blade per propeller every ten years. A breakdown showing the distribution of new and repaired blades among different types of ship is given, together with some tentative average-cost data.

A KaMeWa c.p. propeller of stainless steel (the usual material) is approximately 20% heavier (in water) than a comparable nickel-aluminium bronze f.p. propeller (including shaft end, key, cone, etc.). This might be a problem when considering the use of propellers capable of absorbing over 30,000 h.p. (and requiring very large-diameter tailshafts), because of difficulties which might arise in connection with the design and manufacture of adequate stern-tube bearings and shaft seals, etc.

The article includes a photograph of the KaMeWa c.p. propeller fitted to the *Nuolja* (see Abstract No. 25,144, Mar. 1967).

See also Abstract No. 25,308 (May 1967).

26,782 **Controllable-Pitch Propellers in Large Stern Trawlers.** BENNETT, R., CHAPLIN, P. D., and KERR, N. M. *Trans. Inst. Mar. E.*, **80** (1968), p. 257 (Aug.) [15 pp., 4 ref., 10 tab., 23 graphs, 2 diag.; and Discussion: 8 pp., 4 ref., 4 tab., 3 graphs]

Most modern stern trawlers deep-freeze their catch at sea, and are able to spend a much larger proportion of their time on the fishing grounds than the older type of distant-water trawler which preserved its catch in crushed ice. Because of the need for high speed on passage with a deteriorating catch the propellers of these older trawlers were generally designed for maximum power in the free-running condition. Now that a high free-running speed is not so important, most of the new freezer trawlers have c.p. propellers, which enable the high power required by the freezing plant to be supplied by generators driven by the main engine. A typical machinery layout is shown in a diagram. The Authors set out

to provide design information for the c.p. propeller likely to give the best economic results for such trawlers.

Owing to the lack of propeller data for the towing condition (high thrust at low forward speeds) measured-mile trials were made with two stern trawlers, the 185-ft *Saint Finbarr* and the 212-ft b.p. *Arctic Freebooter*, and were later supplemented by some towing-tank tests on the latter vessel. The instrumentation for the sea trials is described; it included a new type of transducer for measuring propeller torque and thrust. The results were compared, on the basis of plots of K_Q against K_T, with Troost propeller-chart data modified in accordance with a method explained by the Authors. The agreement was sufficiently good to warrant the conclusion that propellers for this type of trawler can be designed on the basis of existing chart information, although further confirmation by similar comparisons is desirable, particularly as regards determination of wake-fraction and thrust-deduction factor.

The Authors show how, by plotting shaft horsepower against pitch-diameter (P/D) ratio, the propeller-chart or trials data can be analysed to show the relation between horizontal pull when towing the gear, and propeller-shaft horsepower, pitch, and r.p.m. A series of curves is obtained, each of which corresponds to a certain pull and towing speed, and it is found that for all values of pull there is an optimum propeller P/D ratio for which the shaft horsepower is a minimum. For the *Arctic Freebooter* this optimum P/D ratio was 0·527 for all pulls and towing speeds within the limits set by the design of the equipment. For this ship, the pulls required in service lie between 7 and 10 tons, which correspond to r.p.m. for minimum power of 180 and 215 respectively. Operation within this range would reduce the power required for towing on the average by 100 h.p. below that required at the average shaft speed now being employed of 250 r.p.m.

The propeller diameter—a variable in which the designer has considerable freedom of choice—has a marked effect on the performance, and the Authors discuss this in relation to two large stern trawlers, 215 ft long b.p., of basically similar design but of different average free-running speeds, viz 14·3 and 12·75 knots. Three possible schemes of propeller operation are considered, namely (*a*) free choice allowed for propeller r.p.m. for the free-running and towing conditions; (*b*) minimum towing r.p.m. limited to two-thirds of maximum free-running r.p.m.; and (*c*) constant-speed operation. At present, (*a*) can be used only in vessels with independent electrical supply; (*b*) represents the present limit set by manufacturers of voltage-control equipment on shaft-driven generators for satisfactory operation. Using propeller-chart data for the Troost B4/55 and B4/70 series, the Authors show that for all three types of operation and for both ships there is a considerable reduction in annual fuel consumption as the propeller diameter is increased from 8 to 13 ft, although the curves are flattening out at the larger diameters.

Allowing for typical catch rates, frozen fish prices, fuel costs, capital costs, and depreciation, the Authors estimate that, for the faster vessel operating with scheme (*b*), a 12-ft diameter propeller would reduce the annual running costs by £2,200 as compared with the smallest propeller in current use (8½ ft) and by £1,600 compared with the average propeller diameter of 9 ft 2 in to 9 ft 4 in, with which 80% of present trawlers are fitted. These figures represent annual fuel savings of 15 and 11%

respectively. For the slower ship the figures are 12 and 10%.

Of the three types of operation, constant speed always leads to higher fuel consumption than the others, but the difference decreases with decreasing free-running speed. On the other hand, with constant-speed operation A.C. power could be generated with alternators driven off the gearbox, and capital and maintenance costs would be considerably less than for D.C. power generation.

In the discussion, the Authors' assumption that freezer trawlers do not need as high a free-running speed as those that store their catch in ice was queried by one of the speakers. A number of points were also raised on the choice of ship for a particular duty and the choice of powers to be installed. In this connection, the Authors provided a graph based on operational research techniques, which showed the economic advantage of increased cold-store capacity up to 45,000 cu ft—the maximum so far studied—and of slower ships in North-West Atlantic fishing.

26,783 **Propeller Maintenance—Propeller Efficiency and Blade Roughness.** BROERSMA, G., and TASSERON, K. *Int. Shipbuild. Progress,* **14** (1967), p. 347 (Sept.) [10 pp., 10 ref., 6 tab., 6 graphs]

If the large-scale form and the clearances of a propeller have not changed during use, the only factor causing its efficiency to deteriorate appears to be the increasing roughness of the blade and boss surfaces. Surface roughness decreases the efficiency in two ways. Compared with the smooth propeller, the drag of the blades is increased and this causes an increase in torque coefficient for the same angle of incidence. Also, the circulation is decreased and this reduces the lift coefficient and the thrust coefficient of the propeller for the same angle of incidence. Relevant parameters (e.g. blade-friction drag coefficient) influenced by the character of the surface are considered in relation to actual propellers and to experimental work by Bussler and by Gutsche (see, respectively, Abstracts No. 11,974, Sept. 1956, and 21,460, June 1964). The roughness parameters used in hydrodynamic research are still those of 30 to 40 years ago. The study of actual surface characteristics, as in the production engineering of high-quality gears, is recommended, together with use of the concept of permissible undulation of a surface. In the case of propeller manufacture, this parameter should be studied in relation to the occurrence of cavitation, with the aim of limiting, or even excluding, the possibility of cavitation due to discontinuities in the centre-of-curvature locus. These new ideas are applied to the analysis of deterioration in propeller efficiency.

An appendix gives some data from roughness measurements on Lips propellers, both in the new condition and after service.

SHIP PERFORMANCE, STABILITY, AND MANOEUVRABILITY

(*See also Abstracts No.* 26,796 *and* 26,818)

26,784 **Sea Spectra Simplified.** MICHEL, W. H. *Marine Technology,* **5** (1968), p. 17 (Jan.) [14 pp., 10 ref., 4 tab., 10 graphs, 6 diag.]

This paper was presented at the April 1967 Meeting of the Soc. N.A.M.E. (Gulf Section).

The paper presents, in a clear and simple form, the fundamentals of statistical sea-spectrum theory and its application to ship behaviour.

Although a " confused sea " (several intermingling seas) is more prevalent in nature, the two-dimensional irregular sea is considered to have maximum effect on a body situated within it.

Recommendations are made regarding the choice of a spectrum formula; that of Bretschneider is considered the most suitable for practical use. Its application is illustrated by calculation of the forces acting on a multicaisson drill rig firmly anchored in 600 ft of water at a draught of 65 ft, in a seaway having a significant wave height of 30 ft and a significant period of 11·5 sec.

STRUCTURAL DESIGN AND ITS APPLICATIONS

26,785 **Applications of the Plastic Theory of Bending [to Ships' Structures].** ROSS, C. T. F. *Shipp. World & Shipb.*, **160** (1967), p. 1533 (Sept.) [3 pp., 3 ref., 12 diag.]

After mentioning the extensive use made in civil engineering of the plastic theory of bending, the Author explains how this theory can be applied to certain components of a ship's structure; he is particularly concerned with the treatment of triangular (hydrostatic) loading. Horne's kinematical approach is used.

The theory followed when determining the plastic moment of resistance and lateral load factor of a vertical bulkhead stiffener (presumed to fail as a three-hinge mechanism) is described, and a worked example is given. If this same vertical stiffener is supported by heavy horizontal cross-members, the possibility of various collapse mechanisms (which are shown diagrammatically) should be considered by the same approach. A simplified rectangular model of a single-deck transverse ship structure is used to illustrate the application of plastic theory to transverse-strength problems, the following collapse mechanisms being considered: (1) Failure in deck beam. (2) Failure in bottom beam. (3) Failure in side frame. These three mechanisms are shown in diagrams. It is noted that, should the side frame be subjected to unsymmetrical loading, collapse may take place by a combination of mechanisms, in which event the calculations for the combined mechanism can be made by a method similar to that suggested by Symonds and Neal.

These examples show the simplicity of application of the plastic theory to some quite difficult problems. It is noted that, if flange or plate buckling is likely to occur, allowances for this must be made in the calculation of the plastic moment of resistance of the section. Although the concept of the plastic theory is based on materials which have a load-extension relationship like that of mild steel, the theory can be applied to any alloys which have sufficient ductility for the formation of plastic hinges. Manufacturing imperfections affect the plastic collapse load, but their effect on plastic theory is much less than on elastic theory.

Experimental verification of the results is needed.

26,786 **Tests on Bracketless Knees.** REDSHAW, S. C., and SHAKIR-KHALIL, H. *Trans. Instn E. Shipb. Scot.*, **110** (1966 67), p. 291 (Pt. 7) (Paper No. 1324) [33 pp., 20 ref., 6 tab., 5 graphs, 17 diag., 9 phot.; and Discussion: 4 pp.]

The Authors give an account of two series of tests sponsored by B.S.R.A.

In the first series, carried out on full-scale specimens in a 50-ton

hydraulic test-machine at the University of Birmingham, the structural behaviour of twelve designs of welded square bracketless knee connections was investigated comparatively. Although the test conditions and the specimens were not fully representative of ship conditions, it was possible to select two designs which were stronger and stiffer than the others; one of these was judged to be the best of the twelve as it was the easier of the two to fabricate. Two other designs were the simplest and made the least exacting demands on fabrication accuracy; one of these two had good load-carrying capacity and was used in the second series of tests, carried out on two portal frames embodying, respectively, six bracketed and six bracketless connections.

The tests on the portal frames were done at Glengarnock. The object was to ascertain how, in a built-up structure representative of that in a hull, the performance of a bracketless knee connection would compare with that of a bracketed one. It was concluded that, for the type of portal used in the tests, the bracketless connection used is the better and more economical solution.

The results of both series of tests are presented in detail and discussed. Information on the test procedures and equipment, drawings of the specimens and portal frames, and photographs of specimen failures, are included.

26,787 **Application of Mathematical Statistics to the Analysis of [Present] Practice in Establishing the Frame Spacings of Merchant Vessels** (in Russian). LAZAREV, V. N. *Trans. Leningrad Shipbuild. Inst.*, No. 54 (1967), p. 43 [6 pp., 3 ref., 5 tab., 6 graphs]

The Author has carried out a statistical analysis of the frame spacings (longitudinal or transverse) of 380 dry-cargo ships and 100 tankers, with the aim of promoting standardisation of hull structures by unifying the frame-spacing requirements of the different Classification Societies. (The existing differences in their requirements are illustrated by a comparative table.) The ships concerned were both Russian and other vessels of various designs constructed during the last fifteen years.

The Author concludes that such unification of frame spacing is desirable, and that analysis of such statistical material creates a practical basis for it.

26,788 **Hardy-Cross Calculations for Hatches with Partly Elastic Support for the Fore-and-Aft Coamings** (in German). SCHWAGER, G. *Hansa*, **104**, No. 24 (1967), p. 2147 (Dec.) [6 pp., 9 ref., 14 tab., 3 graphs, 8 diag.]

26,789 **A Comparison of Measured and Calculated Stresses in Pressure Vessels.** KRAUS, H. *A.S.M.E., Paper No. 66-WA/PVP-5, presented* 27 Nov.–1 Dec. 1966 [4 pp., 8 ref., 10 tab., 1 diag.]

Stresses as calculated by two available computer programs for the analysis of pressure vessels are compared in tables with photoelastic data obtained by other workers for a series of pressure vessels with hemispherical and torispherical heads. One program is based on Love's first approximation to the theory of thin elastic shells, and the other on a higher-order theory of shells. The comparison shows that programs based on either theory can be used with equal confidence in the calculation of stresses in pressure vessels.

26,790 **On the Dynamics of Liquids [Sloshing] in a [Vertical] Cylindrical Tank with a Flexible Bottom.** SIEKMANN, J., and CHANG, S.-C. *Ingen.-Arch.*, **37** (1968), p. 99 (No. 2) [10 pp., 14 ref., 5 tab., 1 diag.]

WELDING AND OTHER METHODS OF CONSTRUCTION

(*See also Abstract No.* 26,808)

26,791 **Weld Testing—Offshore Structures.** TURNER, R. P. *A.S.M.E., Paper No. 66-PET-30, presented* 18–21 Sept. 1966 [6 pp., 2 diag., 6 phot.]

Reasons are given for selecting the shear-wave ultrasonic technique as the most practical means of checking the condition of welds, both above and below water, in offshore structures. The principle of the technique is outlined, and the methods used in applying it to platforms standing in up to 200 ft of water are described and illustrated. The transducer is handled by a trained diver in telephone communication with two experts on the platform, who are in charge of the remaining equipment and interpret and record the signals fed to it from the transducer via a long coaxial cable. The first complete survey of a platform's underwater structural welds was made in 1963. The records can be compared with those made at the builder's yard or during previous *in situ* checks.

SHIPBUILDING (GENERAL)

26,792 **Hull Steel Weight of Dry-Cargo Ships** (in German). SCHNEEKLUTH, H. *Schiff u. Hafen*, **19** (1967), p. 84 (Feb.) [2 pp., 4 graphs]

At the Technische Hochschule, Aachen, a study is being made (with the support of Germanischer Lloyd and others) to evolve a new method, consistent with current GL Rules, for calculating hull steel-weight in the design stage. The method would give hull steel-weight, and its vertical CG, and apply only to dry-cargo ships; it would therefore complement rather than replace the usual methods, which are to some extent based on older Rules. The main object of the work is to get a better understanding of the relationship between hull steel-weight and hull proportions, in the light of current design knowledge and practice; this is necessary for design optimisation.

The Author compares a method resulting from the study with methods proposed by others in this field, and presents a formula for calculating the hull steel-weight of a proposed ship when that of another ship having the same main dimensions but a different block coefficient is known.

See also following Abstract, and Abstract No. 21,661 (July 1964).

26,793 **A New Method for Determining the Steel Weight of Sea-Going Ships** (in German). CARSTENS, H. *Hansa*, **104** (1967), p. 1864 (Special Issue in Nov.) [11 pp., 10 ref., 24 tab., 8 diag.]

After briefly reviewing the work of others in this field (including the study covered by the preceding Abstract), the Author describes his method for estimating the steel weight during the initial planning of a ship design. An early stage in the development of this method has previously been described (see Abstract No. 17,910, Dec. 1961); since then, the method has undergone considerable further development, but the principle of relating weight to surface area (not to volume, as in most other methods)

has been retained throughout. A computer program is available for use with the method.

The development of the method is described; its application is explained in detail, with a numerical example. Its accuracy has been checked in the case of a number of existing ships, and its practicability has been demonstrated. An accuracy of $\pm 2\%$ (of total steel weight) is obtainable, provided the basic information is satisfactory (there must be at least a general-arrangement sketch, and the displacement and type of construction must be known). If, in the course of the design work, the design is modified, new weight calculations can readily be made without having to repeat the whole procedure.

It is mentioned that, although some people may consider that the steel weights estimated by this method are too low, they have, in fact, been achieved.

26,794 **Steels for Ships.** *Brochure issued by the Appleby-Frodingham Steel Co. (published by The United Steel Companies Ltd)*, 1967 [52 pp., 16 tab., 58 graphs, 5 diag.] See also *Shipbuild. Shipp. Rec.*, **111** (1968), p. 87 (18 Jan.) [2 pp., 10 tab., 3 diag.]

This publication contains much general information and numerical data on both mild and high-strength shipbuilding steels supplied by the Appleby-Frodingham Steel Co. The plate grades covered are A, D, E, AH2, DH2, and EH2; the section grades covered are A and AH2. The data include mechanical properties, Charpy values, chemical composition, carbon equivalents, forms of supply, and sizes available.

Also included are some tabulated results of a design study, sponsored by The United Steel Companies and carried out by B.S.R.A., in which the overall economics of using high-strength steels in ship construction were assessed. (These results are also given in the *Shipbuild. Shipp. Rec.* article.) Tables show the principal scantlings, steel weight, steel cost, deadweight, lightweight, and total cost when each of three ship-types incorporates high-strength steels, to various extents, in its hull. These tables, which include the corresponding figures for mild-steel hulls, indicate that: (i) In the case of a 42,500-ton d.w. bulk carrier designed to Lloyd's Rules, using high-strength steel as longitudinal material in the deck results in a saving of £22,000 in total cost as compared with mild steel; the use of high-strength steels as longitudinal material in both the bottom and the deck would be slightly less advantageous than this. (ii) For a 68,000-ton d.w. bulk carrier designed to Det Norske Veritas Rules, and whose dimensions did not allow the full advantage of high-strength steels to be obtained, there would be a saving of £50,000 by using these steels, instead of mild steel, as bottom and deck longitudinal material; the use of high-strength steel for the deck longitudinal material only, or for all the longitudinal and transverse material in the cargo section, would not result in such a large saving as this. (iii) In a 170,000-ton d.w. tanker designed to Lloyd's Rules, there would be a saving of £134,000 by using high-strength steel instead of mild steel for the bottom and deck longitudinal material; using high-strength steel for all the longitudinal and transverse material in the cargo section would save rather less than this.

The savings quoted above assume that fabrication costs for high-strength steels are the same as those for mild steel. The tables also give

figures which assume that the fabrication of high-strength steel costs 25% more than that of mild steel; in this case the savings, though still substantial, would be lower.

Further tables show, for each of the three ship designs, the increases in deadweight and the reductions in capital cost per ton d.w., resulting from using high-strength steels to various extents.

26,795 **Design and Economics of Atlantic Container Ships.** GORGE, J.-N. *Fairplay*, **228** (1968), p. 140 (4 July) [6½ pp., 4 ref., 11 tab., 4 graphs]

There is a tendency among leading shipping companies to order fine-form, fast container ships costing on an average £4 m, about 600 ft long, capable of carrying about 1,000 containers across the Atlantic at 22 knots or more, and requiring about 30,000 s.h.p. The Author examines the advantages or drawbacks of fine forms as against fuller forms such as those of tankers; and also whether the basic investment costs could be reduced by buying Liberty-replacement ships and modifying them if necessary.

He first discusses some general design features of container ships, and derives a formula relating the number (N) of standard I.S.O. containers (20 ft × 8 ft × 8 ft) that such a ship can carry to her length, breadth, depth, and block coefficient (C_b). He then uses this formula, which shows that N is a direct function of C_b, to calculate the number of containers that can be carried by three ships of the same L_{pp} (600 ft), B (95 ft), D (60 ft), and draught (32 ft), but of C_b 0·60 (Ship I), 0·70 (Ship II), and 0·80 (Ship III) respectively. The respective displacements (loaded) are 31,270, 36,480, and 41,690 tons. With engines developing 30,000 s.h.p., the service speeds are 22, 19·5, and 16·75 knots. The same analysis is also applied to Ship IV, a 460-ft b.p. ship, for which B is 64 ft, D 52 ft, draught 28 ft, and block coefficient 0·60. These dimensions are similar to those of the Liberty-replacement ships designed by many major shipbuilders. Her displacement is 14,000 tons and, with 22,000 s.h.p., her service speed 21 knots. N for Ships I, II, III, and IV is found to be 1,054, 1,238, 1,410, and 520, which, to allow for contingencies, the Author reduces to 1,000, 1,170, 1,340, and 500 in the calculations that follow concerning the economics of these ships. For these calculations, certain assumptions are made regarding cost of hull and machinery, cost of containers, hours worked per year (7,920), fuel costs, number of cranes used, etc. It is assumed also that depreciation is 10% for the ship and 20% for the containers, and that two sets of containers are required for each ship. Some of the results of these calculations are as follows:—

Basic costs (£)	*Ship I*	*Ship II*	*Ship III*	*Ship IV*
Hull and machinery	4,000,000	4,250,000	4,500,000	1,800,000
Containers	1,100,000	1,287,000	1,474,000	550,000
Total	5,100,000	5,537,000	5,974,000	2,350,000
Annual costs, (£), total*	1,596,020	1,726,380	1,844,355	819,610
Number of containers carried per year	44,000	46,800	48,240	22,000
Cost per container carried	£36·27	£36·88	£38·23	£37·25

* Including amortisation, depreciation, maintenance, crew, fuel at £6 per ton, port charges, cargo charges, etc.

The figures for costs per container for Ships I, II, and III show the

advantage of fast fine forms for container traffic. To transport the same number of containers that Ship I can carry per year, i.e. 44,000, would cost £26,840 more per annum by Ship II, and £86,240 more by Ship III. Ship I has other advantages as well. With her fewer containers she will be more easily loaded, more likely to operate fully-loaded, and will probably be able to offer better delivery terms—an important consideration in the Atlantic container trade.

The calculations show also that two vessels of Ship IV type can be bought for £400,000 less in basic investment than one of Ship I, and that these two ships can carry the same number of containers per year as Ship I. This saving in basic investment will take more than nine years to be balanced by the increased running costs. On a basis of net present value, this saving is never balanced. Ship IV also possesses in increased measure the other advantages of Ship I over Ships II and III. With two such ships a smaller shipowner could probably maintain a weekly service in each direction across the Atlantic.

26,796 **Preliminary Design of a Catamaran Submarine Rescue Ship (ASR).** MEIER, H. A. *Marine Technology*, **5** (1968), p. 72 (Jan.) [5 pp., 1 tab., 2 graphs, 2 diag.]

This paper was presented at the A.I.A.A./Soc. N.A.M.E. Advance Marine Vehicles Meeting, Norfolk, Virginia, 22–24 May 1967.

The U.S. Navy's decision to design a twin-hull rather than a single-hull vessel to replace existing ASRs was influenced by the handling and support requirements of a new design of rescue submersible. A vessel was required which had the ability to handle a 30-ton rescue submersible (60 tons with entrained water), had a high degree of stability, and was highly manœuvrable at low speed. The catamaran fulfilled all these requirements, with the added advantage of a 40% increase in deck area compared with a single-hull ship of the same displacement. The use of a catamaran also made it possible to handle the submersible amidships between the hulls, where roll and pitch would cause no vertical motion and where there would be some shielding from the seas, thereby reducing the risk of damage to the submersible.

The main particulars of the preliminary design are stated in the paper; however, the final version of the contract design was increased in length. (See also Abstract No. 26,215, Mar. 1968.)

Model tests on a symmetrical and an asymmetrical hull form were carried out. Body plans of the models are given. The principal particulars of both forms are identical:—

Length, b.p.	210 ft
Beam, each hull	24 ft
Draught	18 ft
Displacement, each hull	1,397 tons
Design speed	16 knots
Prismatic coefficient	0·55
LCB	0·502 L aft of FP

Power/speed curves show e.h.p. and s.h.p. for the symmetrical and asymmetrical forms, and a comparison is made with corresponding Taylor-series single-hull and catamaran forms. The effect of hull

separation was insignificant over the practical range of variation; the relevant test results are shown in a graph.

Both hull forms were tested in head and following seas at speeds up to 16 knots and sea states up to 7. Contrary to expectations, vertical magnification of waves passing between the forebodies of the symmetrical hulls was not observed, but there was a very high pile-up of water between these hulls at the after quarter point. This pile-up did not occur with the asymmetrical configuration, and, as there was little to recommend the symmetrical form from the powering standpoint, the asymmetrical form was selected.

Model tests were conducted on the asymmetrical catamaran to determine motion characteristics. The model was tested in irregular unidirectional seaways (at various headings) corresponding to 5 and 15 knots, and to sea states from 5 to 6. The full-scale natural motion periods predicted from the model tests are: heave, 5·35 sec; pitch, 5·35 sec; roll, 6·17 sec.

The propulsion system selected is based on Diesel engines geared to a controllable-pitch propeller in each hull; installed power is 2 × 3,000 s.h.p.

Turning and manœuvring tests were conducted early in the design, with particular emphasis on handling characteristics with the machinery working in one hull only; even under these conditions the ship could be kept on a straight course. In normal operation the vessel was directionally stable, and had good coursekeeping and turning characteristics.

The submersible-handling area is located slightly abaft midships; a large platform-type hoist is installed in the well. The forward superstructure and hull spaces house the command and control spaces and accommodation. The after superstructure and hull spaces house the workshops, etc. The arrangement of spaces in both superstructures is restricted by three transverse structural bulkheads aft and three forward, in addition to the port and starboard longitudinal bulkheads that are extensions upwards of the inboard sides of the hulls.

The hull-girder stresses are extremely low, due to the very low length/depth ratio of the individual hulls. Consequently, hull scantlings are governed by local-strength requirements. For particulars of the structural investigations, see Abstract No. 26,215; a short review of this work is given in the present paper.

26,797 **Numerical Calculation of Areas and Moments [by Computer].** GUSTEREN, L. A. VAN. *Int. Shipbuild. Progress*, **14** (1967), p. 357 (Sept.) [7 pp., 2 ref., 4 diag.]

26,798 ***Tunadal*—Swedish Forest Product Cargo Carrier.** SECRETAN, P. C. *Shipp. World & Shipb.*, **160** (1967), p. 1369 (Aug.) [8 pp., 6 tab., 19 diag., 9 phot.]

The *Tunadal* is the first of three specially-designed ships being built by Lindholmens Varv AB, Gothenburg, for Svenska Cellulosa AB, Sundsvall, and is now in service. These ships will form part of a new integrated distribution system, having two Svenska Cellulosa (SCA) terminals in Sweden and discharge terminals situated in London, Hamburg, and Rotterdam. The main cargoes will be pulp, paper, sawn timber, and

board. The principal particulars are:—

Length, o.a.	153·74 m (504·4 ft)
b.p.	142·1 m (466·3 ft)
Breadth, moulded	20·2 m (66·25 ft)
Depth, moulded	
to upper deck	11·2 m (36·75 ft)
to second deck	8·8 m (29 ft)
Draught, design	7·95 m (26 ft)
Deadweight, corresponding	11,500 tons
Gross tonnage	9,357·57
Block coefficient	0·65
Rectangular bale capacity	13,365 cu m (471,540 cu ft)
Maximum deck-load capacity	1,500 tons
Machinery output	
maximum continuous	8,000 h.p. (metric)
normal service	6,400 h.p. (metric)
Service speed	15·5 knots
Trials speed, max. average	17·02 knots
Classification	Lloyd's Register ✠ 100 A1, ✠ LMC Ice Class 1

The *Tunadal* is an all-aft two-deck vessel of the "all-hatch" type, with poop and forecastle. Two 20-ton Munck gantry cranes travel fore and aft on tracks outboard of the hatch coamings. With both cranes in operation a 7,500-ton cargo can be discharged or loaded in 24 hours. The cranes can be fitted with a pulp-bale clamp which speeds up handling of pulp cargoes; it enables 64 bales to be handled as a single lift.

There are five holds separated by vertically corrugated bulkheads, and narrow wing tanks for water ballast (Nos 1, 2, and 3) or fuel (Nos 4 and 5) in way of them. These wing tanks do not extend above the second deck, where the side spaces serve as access tunnels. The holds are rectangular, with stools in the bottom of Nos 1 and 5 to square off hull curvature. All holds have timber ceilings and sheathing, with drainage space beneath the ceiling. Hydraulically-operated folding steel hatch covers, designed to carry a uniform deck load of 2 tons/sq m (0·18 tons/sq ft) without intermediate support, are fitted to each hold. The weather-deck covers are 17·3 m long by 15·17 m wide (56·8 by 49·8 ft) except at No. 1. Up to 1,500 tons of the 8,500 metric tons which the vessel will normally carry can be deck cargo. The upper tiers of cargo stowed in the holds and tweendecks can be dunnaged by a system of inflatable rubber bags.

Longitudinal framing is used in the deck, top of sides, and double bottoms, with transverse members about every 7 ft; other side framing is transverse. The stern frame is of welded plate, with the sternpost boss and part of the sole piece welded to the shell plating. A streamlined balanced rudder is fitted. The pintle is of forged steel and the lining of stainless steel. The bottom tanks under the engine room are arranged for Diesel and lubricating oil; the others are for water ballast only. Water ballast is also carried in deep tanks forward of No. 1 hold. There is a pipe tunnel (duct keel) in the double bottom.

The accommodation includes three double-berth cabins and a lounge for passengers, on the third poop deck.

Deck machinery, supplied by Thrige, consists of a 75-h.p. windlass with

an A.C. slip-ring motor, and four 8-ton self-tensioning winches. The rotary-vane steering gear is by A/S Porsgrund. The stern tube is oil-lubricated.

The propelling machinery consists of two unidirectional ten-cylinder Lindholmen-built Pielstick PC2V 400 engines, each developing 4,000 b.h.p. at 425 r.p.m. They are equipped to burn 3,500-sec fuel. The output is transmitted through a single-reduction gear fitted with two disc-clutch couplings, which reduces the speed to 130 r.p.m. of the KaMeWa four-bladed controllable-pitch propeller. Combinator-type pitch and engine-speed controls are provided in the wheelhouse and on each bridge wing; control can also be effected from the machinery control room, which is on a flat aft of the main engines and forward of the three Diesel-generator sets (each of which is rated at 450 kVA, and has a Nohab four-stroke five-cylinder turbocharged engine). Control of the main engines is by a Lindholmen pneumatic system, similar in principle to that first installed in the *Yakima Valley* (see Abstract No. 21,173, Mar. 1964). Details of the remaining auxiliary machinery and equipment are given; it includes a waste-heat boiler, a fully-automatic oil-fired boiler, and air-cooled starting compressors. The main ballast pump has a capacity of 300 tons/hr at 25 m (82 ft) head. The valves in the bilge and ballast systems are hydraulically actuated and have remote control from a panel in the engine room.

The article includes a deadweight scale, general-arrangement and machinery-layout drawings, steel plans, structural sections and detail drawings, and tables of hold and tank capacities.

26,799 ***Suffren* — Fast Cargo Vessel for French Owners.** *Shipp. World & Shipb.*, **160** (1967), p. 1538 (Sept.) [7 pp., 1 tab., 6 diag., 9 phot.] See also *Holland Shipbuild.*, **16** (1967), p. 60 (Oct.) [3 pp., 1 diag., 5 phot.], and *Motor Ship*, **48** (1967), p. 239 (Sept.) [3 pp., 3 ref., 2 tab., 6 phot.]

The sister ships *Suffren* and *Rochambeau* are highly automated general-cargo vessels owned by the Compagnie Générale Transatlantique and operating mainly between Le Havre and New York. The *Suffren*, which entered service first (in mid-1967), was built by the Chantiers de l'Atlantique, St Nazaire, and the *Rochambeau* by the Chantiers Navals de la Ciotat, Le Trait. The ships are open shelterdeckers under the new regulations (there are no tonnage openings). They can carry 116 standard 20-ft containers in addition to other general cargo, and can transit the Panama Canal and the St Lawrence Seaway. Their principal particulars are:—

Length, o.a.	149·5 m (490·3 ft)
b.p.	140·25 m (460·1 ft)
Breadth, moulded	20 m (65·6 ft)
Depth, at upper deck	11·6 m (38·1 ft)
Draught, loaded	7·5 m (24·6 ft)
Deadweight	7,500 tons
Tonnage, gross/net	9,848/5,460
Cargo capacity	
bale	15,000 cu m (529,720 cu ft)
refrigerated	620 cu m (21,895 cu ft)
liquid	120,000 l (26,400 gal)
containers (20 × 8 × 8 ft)	116

Machinery output, m.c.r.	2 × 7,090 h.p.
Service speed	19 knots
Trials speed, in ballast	22·8 knots
Complement	27 + 5 passengers
Classification	Bureau Veritas ✠ 1.3/3–L.1.1 A and CP RMC and RMCV-Ice 111-F60

There are four cargo holds forward of the machinery space and one aft; they all have upper and lower tweendecks. The forecastle deck extends some way aft and carries No. 1 weather hatch. A long poop deck carries No. 5 weather hatch and, forward of this, a full-width five-tier superstructure. The front of the superstructure is curved and is raked aft. Twin side-by-side funnels are fitted immediately aft of the navigation bridge. The hull has a bulbous bow, a soft-nosed clipper stem, and a cruiser stern with horn rudder.

There are six independent liquid-cargo tanks below No. 1 lower tweendeck. The refrigerated space in the poop tweendeck of No. 5 hold is divided into four separate compartments, which can be maintained at any temperature between 53·6° and −4° F by direct-expansion plant served by eight Freon 12 compressors. Access to these compartments is through four small poop-deck hatches near the corners of No. 5 main hatch. Deck cargoes can be carried in way of Nos 2, 3, and 4 holds, and fork-lift trucks of three and four tons capacity can operate in the tweendecks and holds respectively. Nos 3 and 4 trunks each have three side-by-side hatches on each deck; to compensate for these wide openings, high-tensile steel has been used for sheer strake, stringer plate, most of the upper deck, upper-deck girders, and strengthening of hatch corners on the main deck. In addition, longitudinal box-girders run under each deck in line with both sides of the centre hatches; they are continuous over most of the ship's length. These girders are connected by pillars at each corner of the centreline hatches.

All hatch covers (27 in all) are of Ermans design and manufacture; those on the weather and main decks are watertight. The covers for the centreline hatches on the weather deck are of the fore-and-aft rolling type and run on raised coamings; the remainder (including Nos 3 and 4 weather-deck side covers) are of the flush-fitting sliding type. The hatch dimensions are stated in the *Shipp. World* and *Motor Ship* articles. Brissoneau & Lotz electric deck cranes can work simultaneously with derrick booms in each trunk. This gear is arranged as follows: two 7-ton booms forward of No. 1 hold; two 5-ton deck cranes between Nos 1 and 2; a portal between Nos 2 and 3, carrying two 10-ton and one 30-ton booms for No. 2 and two Farrel-type "mechanised" booms for No. 3; two 5-ton cranes between Nos 3 and 4; two Farrel-type booms on the superstructure front; two 7-ton booms on the funnels (forward of No. 5), and a 3-ton crane aft of No. 5. The Farrel booms (see also Abstract No. 22,900, Apr. 1965) can be topped and slewed under load; they are rated at 10 tons for normal service and 2·5 tons for burtoning.

Holland Shipbuild. has drawings showing the layout of the cargo-handling gear.

The wheelhouse is the only place where conventional watchkeeping is observed when at sea. Its layout is unusual, a windowed passage being carried right round a central "island". Below the forward windows are two consoles, one each side of the steering pedestal. The starboard

console houses the controls for the main engines, propeller pitch and speed, and the 600-h.p. bow-thrust unit. The port console includes communications equipment, and controls for the windlass and the six self-tensioning winches (all supplied by Brissoneau & Lotz). Navigational aids are grouped in a free-standing console. Another feature is stereophonic reproduction of sounds as " heard " on the forecastle.

Propulsion is by two 16-cylinder SEMT-Pielstick PC2V engines, each developing 7,090 h.p. at 450 r.p.m. These are intended to run on heavy fuel even during manœuvring periods; the fuel is treated by a Sofrance 7036 filtration unit (see Abstract No. 22,570, Jan. 1965). They are coupled through ACB hydraulic clutches and reduction gearing to a single shaft driving a KaMeWa controllable-pitch propeller at 126 r.p.m. A shaft brake is provided.

At sea, electricity is obtained from two 550-kW Brissoneau & Lotz alternators; these are directly driven from the forward ends of the main engines, and are designed to run at 450 r.p.m. Two 550-kW Diesel sets with Pielstick 8PA4 engines (1,200 r.p.m.) are used during manœuvring periods or in port.

Control of the propulsion machinery and the generating plant, in the " manœuvring " and the " at sea " modes, can be carried out through a pneumatic system either from the bridge or from an athwartships control room at the forward end of the engine-room flat. The possible modes of control resemble those of the *Tyro* (see Abstract No. 26,615, Aug. 1968). The control room also houses the main switchboard, the 398-point Saxby data logger, synoptic panel, alarms, programmer, etc. The numerous functions of the automatic machinery-protection system are listed; the system is controlled by a Thomson-Automatismes master panel. Features which help to ensure safe unattended operation include special leak protection for H.P. fuel lines, and fuel-leak detectors.

Steam is provided by a 1·5 tons/hr oil-fired Clayton boiler and a 3 tons/hr double-flow exhaust-gas boiler. Steam production is automatic, and a pressure of 50 to 87 lb/sq in can be maintained.

The *Shipp. World* article includes general-arrangement and machinery-layout drawings, a midship section, and a deadweight scale.

26,800 **The *Glomar Challenger*.** *Maritime Reporter*, **30** (1968), p. 26 (1 July) [2 pp., 1 diag., 4 phot.]

The *Glomar Challenger*, a self-propelled drilling ship, built by the Levingston Shipbuilding Co., Orange, Texas, for Global Marine Inc., was completed in June. She will be used to conduct drilling operations across the Atlantic and Pacific, in depths of 3,000 to 20,000 ft, to obtain long cores of ocean-bed material for the Deep-Sea Drilling Project of the (U.S.) National Science Foundation. Her principal particulars are:—

Length	400 ft
Beam	65 ft
Draught	20 ft
Supplies capacity	6,000 tons
Displacement (fully loaded)	10,500 tons
Height of derrick	142 ft

The *Glomar Challenger* is powered by a Diesel-electric installation for drilling, propulsion, and maintaining position. The main power plant

consists of ten Caterpillar D-398 Series B Diesels, giving a total of 10,000 h.p. intermittently and 8,000 h.p. continuously. One standby unit is provided. Each of the twin propellers is driven by three General Electric 750-h.p. geared electric motors. There are also four Schottel-Nederland S-300-L tunnel thrusters, two in the bow and two in the stern, each developing 17,000 lb of thrust. They are used together with the two main propellers to maintain station while drilling. This is effected by dropping a sonar beacon to the ocean bed when the ship arrives on station and then lowering below the hull four hydrophones which will " home " on to the beacon. Their signals will be fed to a computer which will automatically control the propellers and thrusters so as to maintain position. A Muirhead-Brown gyro-controlled tank stabilising system is installed. The accommodation is located aft.

Auxiliary power is provided by three 500-kW Diesel-alternator sets, having engines of the same type as the main sets. Two Aqua-Chem evaporators can produce upwards of 14,000 gallons of fresh water daily. There is storage capacity for 1,000 barrels of drinkable water and 4,800 barrels of drill water. Two Unit Crane Co. cranes are provided, one of 50 tons capacity and the other of 15 tons. Although position is maintained dynamically on site, there are two 7,000-lb anchors, each fitted with 800 ft of chain and operated by conventional handling equipment.

26,801 **A Large Coastal Tanker to Meet Changing U.K. Demands.** *Motor Ship*, **48** (1968), p. 474 (Jan.) [3 pp., 1 diag., 4 phot.]

The *Wheelsman*, a 4,575-ton d.w. coastal tanker, unusually large for this type of vessel, was built by Clelands Shipbuilding Co. Ltd (now part of the Small Ship Division of the Swan Hunter Group) for C. Rowbotham & Sons (Management) Ltd. She was built to conform to the requirements of the Regent Oil Co. for a vessel of maximum deadweight capacity to enter the port of Shoreham to feed the new Southwick depot on the Portslade Ship Canal. The length was limited to 320 ft, moulded breadth to 47 ft, and the draught to 18·5 ft when carrying a cargo of at least 3,800 tons at 0·72 s.g. Her principal particulars are:—

Length, o.a.	322 ft
b.p.	305 ft
Breadth, moulded	47 ft
Depth, moulded	24 ft
Maximum design load draught	18·5 ft
Maximum load draught on maximum d.w. and summer marks	19·95 ft
Service speed	12 knots
Register, gross	2,897 tons
net	1,536 tons
Cargo capacity	Over 200,000 cu ft

To meet the owners' demand for maximum manœuvrability in comparatively small ports where the use of tugs is to be avoided as far as possible, the ship has a controllable-pitch propeller and a steering gear capable of hard over to hard over (through 130°) in 18 sec. She is all-welded and was built to Lloyd's ✠ 100 A1 Classification for carrying Petroleum Spirit in bulk below 150° F and to B.o.T. rules Class VII, ex tropics. General-arrangement drawings are given.

Her scantlings are greater than Classification requirements, and allow her to discharge cargo while lying aground. She has two longitudinal bulkheads and four transverse bulkheads, and is arranged for the carriage of cargoes of various grades. All cargo tanks and lines were shot-blasted both internally and externally, and coated with Dimetcote D.4. All lines are of 12-in diameter, and the vessel can discharge three cargoes simultaneously. The cargo-pumping equipment, which was designed to ensure a turn-round of 12 hours, comprises three screw-displacement pumps supplied by Stothert & Pitt, each with a capacity of 364 tons/hr of petroleum spirit against a back pressure of 100 lb/sq in. They are driven by three Ruston Paxman 4RPHXZ Diesels, each developing 211 b.h.p. at 1,200 r.p.m.

All deck machinery is of Norwinch manufacture and the Donkin steering gear is of the rotary-vane type, driven by two 6-h.p. hydraulic units.

The main propelling machinery is a unidirectional pressure-charged Ruston 9ATCM Mark II engine developing 2,380 b.h.p. at 600 r.p.m. It is flexibly coupled to an MWD plain reduction gearbox and gives 2,310 continuous s.h.p. at 200 r.p.m. on the shaft of the three-bladed Seffle c.p. propeller. Single-lever control, of Regulateurs Europa manufacture, for the main engine and the propeller is provided on the bridge, and emergency control for both in the engine room.

One of the main design considerations was economy of manning and maintenance. The complement consists of a master, three mates, three engineers, a radio operator, five seamen, two engine-room ratings, and three catering ratings.

INDUSTRIAL AND ECONOMIC INFORMATION

(*See Abstracts No.* 26,837 *and* 26,839)

SHIPYARDS, DOCKS, AND PRODUCTION METHODS

26,802 **Demonstration of Modern Means of Ship Production by Means of the Example of Straking and Nesting.** LEOPOLD, R. *Int. Shipbuild. Progress*, **15** (1968), p. 185 (June) [17 pp., 5 ref., 16 diag.]

Straking is defined as the subdivision of large areas of uniform thickness into smaller pieces of sizes obtainable from a steel mill. Nesting implies the arrangement of such pieces within the boundaries of a standard-size plate so as to produce minimum scrap.

The Author first develops a general-purpose algorithm for generating plate pieces which can be cut from standard-size plates ordered from a steel mill. From data provided by drawings, specifications, etc., the algorithm generates fabrication data which minimise the number of different plate sizes to be ordered, the amount of scrap, and the amount of welding. The algorithm has been programmed for a digital computer, and has been shown by tests to save time and cost and to improve accuracy. The various steps are described in some detail.

The nesting process is based on a new concept which the Author calls " force placement ", and it is applied to the separate pieces of various shapes which have to be cut according to the results of the straking process. This also can be done by computer.

26,803 **Contour Cutting of Large Plates.** *Engineering Materials and Design*, **11** (1968), p. 948 (June) [1 p., 1 diag., 2 phot.]

The British Oxygen Co. Ltd have developed a new form of tractor which can be used to guide a tool, cutter, or marking device by manual or remote control over large areas. One application is therefore the cutting of ships' plates so large that they are beyond the range of normal pantographic machines. Other possible uses are the manœuvring of loads or equipment, such as fork-lift trucks, in confined spaces, over distances of 100 ft or more.

The tractor consists of a body, square in plan form, with four identical caterpillar-like tracks, one along each vertical side. Parallel tracks are driven in pairs, and each track section is equipped with lateral rollers, so that the tracks can move freely in a direction normal to the driven direction. The path of the tractor when both pairs of tracks are driven simultaneously is, therefore, in speed and direction, the vector sum of the speeds and directions of the individual pairs. The tractor does not rotate when changing direction and can negotiate sharp right-angled corners.

A prototype recently on show was 2 ft square. A photograph shows a model under telescopic guidance for profiling plate assemblies up to 100 sq ft in area.

26,804 **Application of Numerical Control to Shipyard Production.** BRAYTON, W. C. *Marine Technology*, **5** (1968), p. 39 (Jan.) [12 pp., 1 ref., 8 diag.]

This paper was presented at the September 1966 meeting of the Soc. N.A.M.E. (Chesapeake Section).

The Author describes some of the problems which the Sparrows Point Yard of the Bethlehem Steel Corporation encountered in the transition from traditional shipyard production methods to a system involving automation by numerical control of cutting and layout operations. This superseded a combination of optical marking and full-scale optical tracing by pantograph-type flame-cutting machines. Numerical control, by means of tape, was already being used with success in drilling, boring, and milling operations.

The advantages of numerical (tape) control of flame-cutting machines over optical systems are: tape control is more accurate; compensating devices for shrinking or stretching of the drawing material are not required; positioning of the heads at the start, end, and between cuts is automatic and faster; the cost of paper tape and its handling is very low in comparison with any type of template; simple geometric outlines can be programmed without making an accurate scale drawing.

The operating specifications for the numerically-controlled machine required included the following. It had to be able to cut and/or lay out marking for two or more similar components simultaneously in either parallel or opposite-hand fashion, on one or more plates. This called for four three-torch heads. It had to be able to accommodate two 12-ft wide plates side by side, and all control information had to be readily obtainable from the existing type of 1 : 10 scale drawing (intended for optical marking). To reduce distortion and cutting cost, the cutting had to be co-ordinated so that two non-parallel edges could be cut simultaneously. This required the machine to accept two different instructions for Y-axis travel with one common X-axis travel.

The machine selected was a Linde CM-70 Telerex cutting machine

controlled by an ESSI three-axis tape director using eight-channel paper tape. The servo-control system was designed to provide the operating features already specified. A full description of the equipment and its operation (including the checking and plotting facilities) is given.

All actions are programmed with the exception of the cutting speed, preheat time, amount of kerf compensation, and bevel angle, which are set manually at the machine by the operator. The ESSI director provides either two- or three-axis continuous-path control. The photoelectric tape reader reads digital information from the tape at a rate of 300 characters per second, one block at a time. Each block is executed before the next block is read, therefore programming must be in operational sequence. The path of travel for each block may be a straight line, a circle, or a parabola. The programming of these curves is illustrated.

The tape preparation facilities consist of two Friden Flexowriters equipped with punches and readers capable of handling both the paper tape and edge-punched cards (on which repetitive cuts are recorded), and a Co-ordinatograph which is used to determine the X and Y co-ordinates from 1 : 10 scale development lines. The Co-ordinatograph is connected to one of the Flexowriters so that the co-ordinate dimensions which show on a visual readout panel may be automatically punched into tape by the Flexowriter on depressing a foot switch. The other Flexowriter is an independent unit used for manual programming. With these facilities it is possible to program a complete ship without a computer. Several computer programs are, however, compatible with the ESSI system and can be applied where known dimensions are available.

Manual programming can be used to advantage where drawings are fully dimensional, and also in the production of standard (repetitive) cuts.

Although it is possible to cut nearly every piece of a ship's plating by numerical control, some practical economic considerations limit the extent of its use.

The paper concludes that it is possible, and practicable, to integrate this sophisticated production technique into an existing production system without serious interruption.

26,805 **The Use and Abuse of Critical Path Scheduling.** WALDRON, A. J. *A.S.M.E., Paper No. 67-PEM-26, presented* 10–12 Apr. 1967 [7 pp., 7 ref., 1 tab., 7 diag.]

Most of the paper is devoted to an explanation of the construction of CPM (Critical Path Method) network diagrams, and of their use in project scheduling and resource allocation to the various activities. The form in which a Critical Path Schedule is printed out by a computer is indicated. A partial presentation which is gaining ground is the Time Scale Chart; this is an excellent control tool, although it usually displays only " Early Start " and " Early Finish " dates for each activity; full project control also needs the " Latest Start " and " Latest Finish " dates for non-critical activities, given in the complete Schedule.

The Author considers that, in many cases, users have failed to benefit from CPM owing to certain " abuses " which he proceeds to discuss. The principal abuses are: (*a*) " Massivity," i.e. artificial requirements for an unnecessarily large amount of input data (number of activities) involving an unreasonable " level of indenture " (fineness of detail), and for numerous copies of the output data arranged in various ways.

(*b*) " Periodicity," i.e. the requirement for fortnightly, monthly, or quarterly updating; the technique itself indicates those few and irregularly-spaced " milestones " at which a new schedule should be obtained, and " periodicity " results from a failure to appreciate the difference between reporting and control. (*c*) " Rigidity," which means that, once a schedule has been constructed, deviations in sequences and durations are not allowed unless the entire diagram is redrawn, approval obtained, and a new schedule constructed; in most cases this is unnecessary.

Another mistake is the preparation of a separate CPM schedule for the initial phases of a project; only by chance can this show the true critical path, and the resources expended on it could probably be put to better use.

The Author regards CPM, when properly used, as a simple and flexible information model, generating useful and accurate information that is timely and explicit. It is, however, dependent on the quality of the input, and the sequences used in constructing the diagram are often not the only possible ones.

26,806 **Optimum Conditions for Blast Cleaning of Steel Plate.** REMMELTS, J. *Int. Shipbuild. Progress*, **14** (1967), p. 365 (Sept.) [13 pp., 1 ref., 13 tab., 16 graphs]

Investigations are described in which a laboratory set-up simulated practical conditions of blast-cleaning steel plate covered with mill-scale or light rust; the equipment was of the dry air-blast, not of the centrifugal, type. The principal abrasive used was cut steel wire. The results are presented in detail and discussed.

The rate of cleaning is found to increase when more abrasive per unit of time is discharged. This means that high pressure, a large nozzle, and a wide valve for metering the abrasive flow are advantageous, but the actual settings of these variables should be mutually adjusted. The optimum blasting angle for removing mill-scale is about 45°, and the optimum nozzle-to-work distance 55–75 cm (21·7 in – 29·5 in). The smaller the abrasive grains, the quicker the cleaning process. Air consumption decreases as, under otherwise constant conditions, the abrasive transport is increased. High cleaning rates are advantageous, because they lead to lower costs per unit area for labour and power with little increase in costs for abrasive.

MATERIALS: STRENGTH, TESTING, AND USE

(*See also Abstracts Nos.* 26,794 *and* 26,833)

26,807 **Fatigue Properties of Nonferrous Alloys for Heat Exchangers, Pumps, and Piping.** GROSS, M. R., and SCHWAB, R. C. *A.S.M.E., Paper No. 66–WA/PVP–6, presented* 27 Nov.–1 Dec. 1966 [7 pp., 2 ref., 2 tab., 13 graphs, 3 diag.]

The fatigue behaviour of 13 copper-base and nickel-base alloys used for corrosion-resistant heat exchangers, pumps, and piping systems was investigated over a broad life spectrum of 100 to 100 million cycles. High-cycle tests were of rotating bending type; low-cycle tests were made on flat specimens in alternating flexure. Most tests were made in air, but in some the specimen was continuously wetted with brackish water. The results are presented, mainly as S–N curves. It is concluded that wrought Monel and forged Ni-Al bronze have the highest fatigue strengths,

whereas gun metal and valve bronze have the lowest. The effect of the salt water on fatigue performance was not found to be very significant. The use of Langer's equation to predict stress-cycle relationships gave satisfactory results for wrought alloys, but appeared to be too conservative for cast alloys.

BOILERS AND STEAM DISTRIBUTION

26,808 **Tube Expansion and Welding by Explosives.** MOTTRAM, R. A., and ANDREW, R. V. *Engr*, **224** (1967), p. 402 (29 Sept.) [1 p., 4 ref., 1 phot.]

The Author discusses the use of explosives for expanding and or welding heat-exchanger tubes to tube-plates in the light of the experience acquired by his firm (Imperial Chemical Industries Ltd, Billingham). The technique has been used at Billingham since 1965, in situations where emergency repairs were necessary or leaks were detected during pressure-testing. The joint obtained is usually a purely mechanical one, but a slight modification sometimes used results in explosive welding over part of the joint.

The explosive method has proved much superior to conventional techniques in the following situations: Repairs where access is poor. Joints between dissimilar metals. Welding of external tube cladding to the tube plate (expansion and welding are done in a single operation). Expansion of composite tubes incorporating a soft inner lining (roller expansion greatly reduces the thickness of such a lining). Making good leaky joints discovered during pressure testing or in service (this can be done quickly even when the shell side is under pressure—in one case 1,500 lb/sq in of water was involved). Repair expansions in holes which are oversize or have a bad finish.

The training of personnel in the explosive technique is easier than that for roller expansion or welding; also, the laboratory background work required to establish the best procedure is simpler and less costly than is the case for a welding procedure. It is not easy to cost accurately the types of job so far carried out. A true comparison between the explosive and other techniques must take account of the large reductions in down-time, and of the fact that in many cases less work on the tube-plate is necessary. At present, there is a clear economic advantage in certain situations. Improvements in the technique may make it strongly competitive: it may also permit greater freedom in heat-exchanger design.

DIESEL AND OTHER I.C. ENGINES

26,809 **Application of Pulse-Converters to Two-Stroke Diesel Engines.** JANOTA, M. S., TAYLOR, D. H., and WATSON, N. *Gas and Oil Power*, **64** (1968), p. 198 (Aug.) [7 pp., 4 ref., 1 tab., 5 diag., 14 graphs]

To ensure good scavenging and efficient use of the exhaust-gas energy, turbocharged Diesel engines working at b.m.e.p. up to 250 lb/sq in (four-stroke) and up to 200 lb/sq in (two-stroke) should be operated on the pulse system. However, at higher b.m.e.p. values, operation on the constant-pressure system gives better specific fuel consumption.

Most present-day engines operate either below these ratings or for long periods on part load. Hence they require the pulse system; but the

turbocharger turbine then suffers a loss of efficiency due to partial-admission losses and to the pulsating flow through it. It is therefore desirable to devise a system that satisfies the engine requirements and also provides full-admission flow to the turbine. Three-cylinder exhaust grouping almost achieves this condition, but one- or two-cylinder exhaust grouping does not. In such cases the performance of four-stroke engines can be substantially improved by connecting the different branches of the exhaust manifold before the turbine to a special junction called a pulse-converter.

The efficiency of this arrangement has been established for the four-stroke engine by extensive experimental work, but there is no published information on the application of pulse-converters to two-stroke engines. To provide such information, and to elucidate the characteristics of the unsteady flow of the gas in the pulse-converter, the work described in this paper was undertaken jointly at Queen Mary College, London, and Ruston and Hornsby, Ltd, Lincoln

The performance of engines with less than three-cylinder exhaust grouping is first discussed. A typical example of three-cylinder grouping is provided by the six-cylinder engine; the performance of four-stroke six-cylinder engines is often better than that of multi-cylinder four-stroke engines with other numbers of cylinders, in which the turbine has to operate under partial admission. Despite attempts to reduce the losses in the turbine under these conditions, its efficiency is still some 7 to 12% less for partial than for full admission.

In engines fitted with a pulse-converter, the exhaust manifold arrangement designed for maximum pulse-energy utilisation is normally retained; and the different branches of the manifold are led through special nozzles into a common mixing pipe with a diffuser at the end before entry to the turbine. When properly designed, this pulse-converter allows the flows in the different branches to interact so as to produce the necessary pulse action in the various cylinders, but full admission is provided for the turbine.

The Authors explain the basic principles of the device, and then describe the experiments they carried out, firstly to develop an effective pulse-converter of simple design, and then to investigate its effect on the performance of two two-stroke engines—a four-cylinder Foden high-speed automotive engine, type FD4 Mk. V, and a medium-speed four-cylinder Ruston 4AO. For the development of the pulse-converter itself, an experimental pulse generator was built to explore the effects of variations in areas, mixing section, diffuser, and plenum chamber. The bifurcated type of pulse-converter was preferred to the concentric type, which was not investigated; and the simple design finally tested on the Foden and Ruston engines had neither a mixing length nor a plenum chamber.

These tests showed that the pulse-converter considerably improved the performance of the two engines over a wide operating range. The performance of the Foden engine (798 cc per cylinder) was made to approach that of the six-cylinder version, the specific fuel consumption at 130 lb/sq in b.m.e.p. being reduced from 0·4 lb/b.h.p.-hr for the standard design to 0·38 lb/b.h.p.-hr with the pulse-converter. The turbine admission conditions were also improved. For the larger Ruston engine (2,972 cc per cylinder) graphs show that the specific fuel consumption, air-flow rate, boost pressure, and exhaust temperature were all improved

as compared with the performance without the pulse-converter and also with the performance of the three-cylinder version of the engine.

The Authors state that the work reported in this paper covers only the initial stage of the investigation of the application of the pulse-converter to two-stroke engines.

26,810 **[Theoretical and Experimental] Study on Governing Factors for Cylinder-Head Wall Temperature of High-Speed Diesel Engines.** NUMATA, T., and HYUGA, M. *Mitsubishi Heavy Industries Technical Review*, **4** (1967), p. 172 (No. 3) [7 pp., 4 ref., 3 tab., 17 graphs, 5 diag.]

26,811 **The Case for the Two-Stage Diesel Cycle.** POLAK, P. *Engr*, **226** (1968), p. 330 (30 Aug.) [2 pp., 3 ref., 3 diag., 1 graph]

The Author discusses the concept of the two-stage compound Diesel engine, with the expansion beginning in relatively small high-pressure cylinders and continuing in larger low-pressure cylinders. Various possible cycles and layouts are considered. It is suggested that the mechanical output of the high-pressure cylinders should be used entirely for supercharging (" gas-generator " section). It is stated that calculations for some individual cases have indicated brake thermal efficiencies of up to 55%.

POWER TRANSMISSION

26,812 **Geislinger Couplings for Ship Propulsion Installations** (in German). GEISLINGER, L. *M.T.Z.*, **28** (1967), p. 444 (Nov.) [4 pp., 4 ref., 7 graphs, 2 diag., 2 phot.]

The Author gives information on Geislinger couplings additional to that in the articles summarised in Abstracts No. 25,928 (Nov. 1967) and 17,627 (Aug. 1961). Development work on these couplings over the past few years, the test equipment used, and the principles on which the couplings are based, are described in some detail. It is mentioned that their high damping enables a smaller flywheel to be used; for four-stroke engines with more than six cylinders per row, the flywheel can be dispensed with altogether.

By the middle of 1967 over 1,000 Geislinger couplings had been delivered. Experience has shown that the coupling can be expected to run for at least 60,000 hr without needing the renewal of any main components. However, the O-rings should be changed during major inspections prescribed by the Classification Societies (in general, every 20,000 hr).

Information is given on the vibrational and other characteristics of the Geislinger-coupling installation in Ishikawajima Harima Heavy Industries' Liberty-ship replacement, the " Freedom " class (see Abstract No. 26,087, Jan. 1968). Like the 12-cylinder V-form Pielstick engine in these ships, the coupling used is built by I.H.I. The engine (5,130 h.p. at 500 r.p.m.) is geared, and drives a fixed-pitch propeller; no flywheel or vibration-damper is fitted. Brief information is also given on some other Pielstick-engined installations in which Geislinger couplings are used (e.g. the twin-screw *Black Watch/Jupiter* see Abstract No. 24,776, Nov. 1966).

26,813 **The Voith Turbo-Coupling for Ship Propulsion Installations** (in German). BÖNSCH, G. *M.T.Z.*, **28** (1967), p. 448 (Nov.) [4 pp., 4 ref., 2 graphs, 2 diag., 2 phot.]

The Voith turbo-coupling is a newly-developed hydraulic coupling intended for fitting between the engine and the reduction gearing in geared Diesel propulsion installations; it is particularly suitable for highly-supercharged multi-engine installations. A feature of the coupling is that it provides two different relationships between torque and slip, according to whether the coupling has a full charge or a pre-set part-charge of oil; a change-over control is incorporated. In either condition of charge, the coupling can be rapidly emptied and rapidly filled or part-filled again, so that it can be used as a clutch. In the part-charged condition, the slip is higher and remains substantially constant over the whole operating range of r.p.m.; in this condition it protects the engine from overloading during manœuvring and permits very slow running. Although the oil remains at the pre-set level there is, in fact, a flow of oil for cooling purposes.

The Author describes, in some detail, the construction of the coupling and the principles on which it is based. The first of these couplings were designed for 5,100 h.p. at 475 r.p.m., and the rotor diameter was 2,200 mm (7·2 ft); the part-charge was pre-set to give 25% slip. Two such couplings were installed in the twin-screw ferry *Nils Holgersson*, which entered service in June 1967.

When the coupling has a full charge, i.e. when not manœuvring, its efficiency is about the same as that of other types of coupling (about 97–98%). In the part-charged condition the lower efficiency is of minor importance. The cost of the coupling is less than that of some comparable couplings. The Voith coupling can be supplied without the part-charge arrangement, and can then be used as a simple clutch-coupling.

26,814 **A Criterion for Selecting, for Calculation Purposes, the Critical [Most Highly Stressed] Sections of Intermediate and Propeller Shafting** (in Russian). LUKYANOV, I. S. *Trans. Leningrad Shipbuild. Inst.*, No. 54 (1967), p. 81 [8 pp., 11 ref., 3 tab., 1 graph, 1 diag.]

LAYOUT AND INSTALLATION

26,815 **Vickers' Pipework Fabrication Techniques.** JACKSON, P. B. M. *Fairplay*, **228** (1968), p. 62 (25 July) [2 pp., 1 tab., 2 phot.]

A development programme in which the Vickers Shipbuilding Group's Barrow shipyard has been engaged had for its object the rationalisation and improvement of pipework fabrication. The resulting techniques, which the Author describes, are now established on a production basis, and have led to fewer pipework failures in service, with consequent reduction of maintenance and repair work, together with reductions of weight, size, and installation time.

The main effort was devoted to the elimination of as many joints and castings as possible, since these have always been the greatest source of trouble in manufacture, installation, and service. To replace them, three main techniques were developed, namely, tight-radius bending, extrusion of T-pieces, and swaging of concentric reducers. Each of these is described.

Tight-radius bending can replace many of the forged elbows and brazed fittings often used in conventional pipework systems. The shipyard has standardised on a centre-line radius of 2 × o.d. for these tight-radius bends up to 3·5-in o.d. This limiting size is now being extended to 7·5-in o.d. The materials covered by the process are stainless steel and monel (from 0·84-in o.d. to 2·375-in o.d.), and copper, copper-nickel, copper-nickel-iron, and mild steel (from 0·84-in o.d. to 7·464-in o.d.). The saving in weight as compared with forged elbows and brazed fittings may be as much as 50%. Tests have shown that the method of fabrication, which avoids the usual difficulties associated with tight-bending of ovality and reductions of wall thickness on the outside of the bend, does not impair the fatigue life.

Extruded T-pieces can at present be supplied in copper-nickel from 2·128-in o.d. × 0·116-in wall thickness to 7·464-in o.d. × 0·35-in wall thickness. With this material, T-pieces with greater wall thickness (e.g. 2·875-in o.d. × 0·487-in) cannot at present be extruded, but with copper, copper-nickel-iron, and aluminium-brass, wall thickness imposes no restriction. Here again, fatigue strength is unimpaired.

Concentric reducers, in which a length of pipe is swaged down to give a lower diameter at one end, reduce the number of flanges and welded joints. Only a joint at the reduced end is required, since the larger end is merely the continuation of the pipe from which the reducer has been produced. Copper-nickel tube has been successfully swaged from 7·464-in o.d. × 0·35-in wall thickness to 2·128-in o.d. × 0·116-in wall thickness.

Photographs show (*a*) a cast elbow weighing 36½ lb and an equivalent tight 2-d bend weighing 19 lb, and (*b*) an extruded reducing T-piece weighing 65 lb and the cast reducing T-piece weighing 124½ lb which it replaces.

26,816 **The Design and Fabrication of Pipework without Drawings, by Digital Computer and Models.** REDDING, R. J. *Institution of Chemical Engineers*, paper from *Proceedings of Symposium on Cost Reduction in the Design Phase*, 16–17 June 1966, p. 50 [5 pp., 4 ref., 1 tab., 1 diag., 3 phot.]

After a brief analysis of the conventional use of drawings in the design and fabrication of chemical-plant pipework, the Author describes how computers and scale models can be of considerable assistance in these tasks. These new techniques can make current drawing procedures largely redundant.

Starting with the process flowsheet and the dimensions and positions of the major items in the plant, data can be derived for the terminal points, in space, of each pipe. The rough outline of the route for each pipe can then be decided, and the details fed into a digital computer whose program embodies the rules of good pipework practice. The exact route is decided by the computer; if full details of the fabrication method (such as bending-machine program and details of fittings) are now fed in, the computer can provide an output complete in all details necessary for the manufacture of each pipe run, together with quantities for ordering purposes. Considerable extension of this output is possible.

An important advantage is that the computer is used in a data-processing mode in which all the original and derived data are " fluid " right up to

the time that the final fully-detailed print-out is needed for actual use in construction. Delaying the final print-out in this way means that the print-out procedure should take as little time as possible. The computer output should preferably employ a line printer to avoid the time and expense involved in special draughting machines.

To overcome the objections to such a " blind " procedure, a pipe-modelling machine can be used which quickly produces three-dimensional scale-models of the piping under punched-tape control. Such a machine is briefly described; it produces (by hot-bending) polystyrene pipe-models which are assembled to form a model of the plant. The machine also punches eight-track tape with the input information, which, when merged with the pipe designation, pipe size, and material specifications, becomes the specific input to the computer. The output of the computation includes a revised punched tape which is fed back, off-line, to the pipe-modelling machine to provide a three-dimensional replica of the piping which has been finally decided by the procedure. The tape may be re-run to produce piping for extra models of the plant and to provide pipe-models for manufacturing the pipes.

The practical advantages of these procedures in fulfilling a chemical-plant contract are discussed. Details are given of the time-saving obtained when designing £1 million worth of pipework for a £4 million plant; only 118 man-days and 2½ hr computer time would be needed (assuming the programs were available), as against 5,000 man-days by present " manual " methods.

There is a brief discussion on sub-routines and optimisation.

A short appendix describes the use of the computer procedure in the design and manufacture of tubing for a power-station boiler.

26,817 **Piping Layout Rationalisation by Means of Design Models** (in German). KAYSER, P. *Hansa*, **104**, No. 24 (1967), p. 2126 (Dec.) [3 pp., 1 diag., 2 phot.]

After discussing the problems which shipyards experience in the design, manufacture, and installation of pipe layouts, with particular reference to engine-room piping, the Author describes how design models can alleviate these difficulties.

The techniques suggested by the Author are explained in some detail. A simple but accurately-scaled 1 : 10 or 1 : 20 Plexiglas model, representing the decks and bulkheads in the part of the ship concerned, is constructed as soon as sufficient information is available. This bare model is made up of detachable sections which, when the model machinery and associated equipment and piping are installed, present views corresponding roughly to what would have been cross-sectional views if conventional drawings had been prepared. Simple model pieces, made of rigid foam material, are used for the various items of machinery. The model is built up on the basis of established design data, as far as these may be available, together with a trial-and-error procedure. Commercially available model pieces are used for the pipes and fittings.

On completion of the model, a permanent record of it must be made. This can be done by making isometric dimensioned sketches of the various pipe layouts; these sketches, which need not be true to scale, are also used as working drawings. When ready for use, the model should be kept near the workshops concerned; these methods should enable up

to two-thirds of the pipes to be made in the workshops. Movement of the model from place to place can be avoided by the use of photographs of the model and of its sections; as the various items are differently coloured, colour photography can be used to advantage. The photographs can be used in conjunction with the isometric sketches; for some work they can be used alone. Any modifications found necessary during the work must be notified to the design office so that they can be embodied in the model and other appropriate action taken.

The Author gives further information on these techniques and their economic and other advantages. Some recommendations are included for the period during which they are being introduced.

MARINE POWER INSTALLATIONS (GENERAL)

26,818 **The Practical Application of Computers in Marine Engineering.** GOODWIN, A. J. H., IRVINE, J. H., and FORREST, J. *Trans. Inst. Mar. E.*, **80** (1968), p. 209 (July) [19 pp., 9 ref., 3 tab., 17 graphs, 4 diag.; and Discussion: 14 pp., 12 ref., 2 tab., 11 graphs, 1 diag.]

This paper, based on the experience of the Yarrow-Admiralty Research Department, is intended to give a broad description of the part computers can play in marine engineering design. It describes the difference between digital and analogue computers, their operation and use, and the type of problem each is best suited to solving.

The main advantage in the application of computers lies in the speed of calculation. However, this does not mean that the designer's work is reduced. In some cases the use of computers will increase it, but, at the same time, his work is made many times more effective in securing good and economical design.

Digital computers are better equipped to cope with steady-state design problems; the paper explains how they can be used in the solution of problems in pipe stressing, shafting alignment, steam-turbine design, gas-turbine performance, and flexible mountings for engines. Brief reference is also made to a number of other problems for which digital-computer solutions prove useful and economical; these are: boiler design, steam-cycle heat balances, gearing design, bearing performance, scoop design and performance, and condenser and heat-exchanger performance.

The analogue computer, on the other hand, is more suitable for dynamic performance and control problems with physical time as the independent variable. With this type of computer it is better if the designer himself uses or stands by the machine to gain the full insight available. The examples chosen to illustrate the effectiveness of the application of this type of computer are: complete steam-turbine propulsion-plant dynamics (see also Abstract No. 25,442, June 1967), the dynamic performance and control of an auxiliary exhaust range associated with such a plant, condenser level control, and the transient (manœuvring) characteristics of Diesel-electric propulsion plants. Non-dynamic investigations have also been carried out; for instance, the analysis of the tension on an oceanographic trawl winch.

The paper also describes stopping-manœuvre computations, and presents the results of computer analyses of such transient conditions in

some detail. A digital computer was used, this being considered preferable for accurate analysis of well-known systems. (An analogue computer is recommended for exploratory work on new systems.) Comparisons are made with ship-trials results. The cases investigated cover large and small warships in which thrust reversal is achieved by reversing turbines, reversing gears, controllable-pitch propellers, or turbo-electric drive; also merchant-ship thrust reversal by direct-reversing direct-coupled Diesel engines, direct-reversing geared medium-speed Diesels with pneumatic friction clutches, and controllable-pitch propellers. There are numerous graphs of emergency-stopping characteristics. In discussing these data, arguments are put forward to show that best results are obtained by using the maximum amount of astern power available as quickly as possible, rather than by running the shaft slowly ahead or holding it stopped.

The discussion and the Authors' reply contain further graphs concerning thrust reversal and ship manœuvring.

HEAT TRANSFER AND INSULATION

(*See also Abstract No.* 26,808)

26,819 **The Plate Heat Exchanger.** HARGIS, A. M., BECKMANN, A. T., and LOIACONO, J. *A.S.M.E., Paper No. 66-PET-21, presented* 18–21 Sept. 1966 [8 pp., 17 ref., 1 graph, 8 diag., 5 phot.]

The principles and history of the plate heat exchanger, together with its characteristics and industrial applications, are reviewed. The available design information is also reviewed, various empirical relations for heat-transfer coefficients and pressure-drops being quoted. The favourable heat-transfer coefficients obtainable without excessive pressure drops indicate smaller heating-surface requirements for a given duty than in the case of a shell-and-tube exchanger. Gasket materials such as compressed asbestos permit operation at temperatures up to 500° F. For exchangers having internal support (by plate contact) 300 lb/sq in is a standard design pressure.

26,820 **The Presentation of Heat-Transfer and Friction-Factor Data for Heat Exchanger Design.** SMITH, J. L. *A.S.M.E., Paper No. 66-WA/HT-59, presented* 27 Nov.–1 Dec. 1966 [9 pp., 3 ref., 1 tab., 5 graphs]

The utility of considering a heat exchanger in terms of overall dimensionless passage variables, rather than in terms of unit-area variables, is examined. It is shown that the information usually given as friction-factor and Nusselt-number correlations can be presented in terms of four dimensionless heat-exchanger passage variables. The number of transfer units of a passage is presented as a function of a flow-per-unit-surface number and a pressure-loss number. A dimensionless method is developed for comparing the performance of different surfaces independently of the geometric scale. The influence of fin efficiency is included in the passage correlations by introducing the ratio of the conductivities of the fluid and the metal. A direct method is developed for the design of a heat exchanger to satisfy specified thermal and pressure-loss requirements. The relations between exchanger surface, volume, length, effectiveness, and pressure drop are directly applicable to system optimisation problems.

26,821 **Nomograph gives Batch Heating, Cooling Variables.** CAPLAN, F. *Heat. Pip. Air Condit.*, **40** (1968), p. 149 (Feb.) [2 pp., 3 ref., 1 graph, 2 diag.]

The article considers the heating and or cooling of a quantity (" batch ") of liquid in a tank by a coil in the tank or by a jacket around it. Due to the constantly changing log mean temperature difference involved, the usual heat-exchange equations do not apply. Solutions to this problem are already available; they assume that the overall heat-transfer coefficient, the weight of the liquid, and its specific heat are all constant, that the temperature of the liquid is kept uniform by agitation, and that heat losses are negligible. To aid computations the Author gives a nomogram, based on the resulting equations, which permits determination of the initial or final batch temperature, the temperatures of the heating or cooling medium, the batch weight, the overall coefficient of heat transfer, the heating or cooling time, or the heat transfer area, when the other variables are known. The use of the nomogram is illustrated by three examples.

26,822 **[Theory of] the Dynamics of Heat Transfer from a Heating Element to a Liquid [Inside a Massive Container]** (in Russian). GALICHSKII, A. K., YENAZIN, V. A., and STEPANOV, K. V. *Sudostroenie*, No. 3 (1968), p. 26 (Mar.) [3 pp., 4 ref., 1 graph, 1 diag.]

AIR CONDITIONING, VENTILATION, AND REFRIGERATION

26,823 **Ventilation, Heating, and Air-Conditioning Systems in Merchant Ships** (in German). KLEPPER, H. *Hansa*, **104**, No. 20 (1967), p. 1735 (Oct.) [8 pp., 6 tab., 9 graphs, 1 diag.]

A detailed explanation is presented of the fundamentals of shipboard ventilation, space heating, and air conditioning.

The general capabilities, limitations, and characteristics of the different systems (i.e. ventilation systems, air-conditioning systems, etc.) are tabulated, and examples are given of the application of psychrometric charts to the systems. Among the matters dealt with are: external air conditions, and exposure to the sun, as design considerations for ships operating in specific areas and those liable to operate anywhere; type of ventilation (mechanical or natural), temperature rise, and air-changes per hour, applicable to the various spaces in the ship; importance of amount of external air admitted to system in unit time per person; " comfort " charts, and factors to be considered when selecting the desired temperature, humidity, and other conditions for the various spaces. The article includes much quantitative information, and there is a graphical representation of the desired differences, in enthalpy and in temperature, between the outside and the inside air.

26,824 **Ventilation of Ships' Refrigerated Holds** (in German). EKELUND, B. *Hansa*, **104**, No. 20 (1967), p. 1747 (Oct.) [2 pp., 4 ref., 1 diag., 2 phot.]

The holds with which this article is concerned are those in which fruit is carried, at a temperature a few degrees above freezing-point. Owing to the gases evolved, efficient ventilation of these spaces is essential if the fruit is to be kept in good condition. Shippers usually specify a maximum concentration for the CO_2 in the hold; this can be readily determined and

is a convenient criterion. The article includes a formula for use in determining the amount of ventilation necessary.

The requirements in a hold ventilation system for these cargoes, and the shortcomings of low-velocity systems, are discussed.

An illustrated description is given of the AB Svenska Fläktfabriken high-velocity system. An advantage of this system is its easy operation; all adjustment and control for the different holds is done from a central position. Another advantage is that the ship operator can always give the shipper a guarantee as to the amount of ventilation supplied. The system is arranged so that air from one hold cannot contaminate the air in another; further, the amount of ventilation is independent of wind conditions and of the speed of the circulating fans. The pressure and suction sides are regulated by the same control lever, and are thus kept in balance. The system has Det Norske Veritas approval, and has been installed in numerous ships. See also Abstract No. 25,148 (Mar. 1967).

26,825 **The Use of Ozone in the Refrigerated Spaces and Provision Rooms of Sea-Going Ships** (in German). BLUME, H. *Hansa*, **104**, No. 20 (1967), p. 1749 (Oct.) [2 pp., 2 ref., 4 diag.]

26,826 **A Simple Method of Measuring Flow Rate in Ventilation Systems** (in German). KÖNIG, G. *Schiffbautechnik*, **18** (1968), p. 382 (July) [2 pp., 4 ref., 1 tab., 1 diag., 1 phot.]

AUXILIARY EQUIPMENT AND MACHINERY

(*See also Abstract No.* 26,830)

26,827 **Marine Electric Power Plants with Waste-Heat Turbo-Alternators and Shaft Generators** (in Russian). RUDENKO, E. P. *Sudostroenie*, No. 3 (1968), p. 43 (Mar.) [3 pp., 2 graphs, 1 diag.]

In some motor ships the main source of A.C. power is a turbo-alternator which uses the converted energy of the main Diesel's exhaust gases, or a shaft generator whose speed is stabilised to keep the output frequency constant. This article analyses the energy characteristics of these generating systems and, in particular, examines the dependence of power at the terminals of turbo-alternators and shaft generators on the working conditions of the main Diesel.

The Author suggests that a combined generating plant, where the shaft generator works in parallel and simultaneously with the turbo-alternator, is more rational, and that, despite certain technical difficulties involved in the use of a waste-heat boiler and turbine and a frequency-stabilising system at the output of the shaft generator, the combined plant has many advantages.

The advantages discussed are: greater efficiency due to the full use of the "free" energy of the exhaust gases and the lower specific fuel consumption of the main Diesel in comparison with the auxiliary Diesels; the fuller loading of the main Diesel (through the availability of the shaft generator); and the absence of any break in the generating process (when there is an automatic starting of a reserve source) due to the thermal inertia of the exhaust-gas boiler. The rational selection of power ratings and distribution of loading between the two generators is discussed at some length.

26,828 **High-Head Single-Stage Centrifugal Pumps for Hydrocarbon Services.** Cody, D. J. *A.S.M.E., Paper No. 66-PET 3, presented* 18–21 Sept. 1966 [4 pp., 5 graphs, 3 diag.]

The " partial-emission " centrifugal pump differs from the conventional " full-emission " type in having a straight-bladed impeller and a uniform small peripheral clearance except at the tangential outlet, which leads into a divergent cone. A single-stage partial-emission pump geared up to very high speed (of the order of 10,000 r.p.m.) is not only smaller, lighter, and simpler to construct than a single-stage conventional pump for the same duty, but is more efficient over a practically important range of specific speeds. It is also more reliable, and easier and cheaper to install and maintain. The paper describes an electrically-driven pump of this kind developed by the Sundstrand Corporation, Denver, Colorado, for hydrocarbon pipeline services. Maximum use is made of standardised interchangeable components both to facilitate quantity production and (by appropriate selection from ranges of components) to adapt each individual pump to the user's requirements.

AUTOMATION, INSTRUMENTS, AND CONTROL DEVICES

(*See also Abstracts No.* 26,780 *and* 26,799)

26,829 **Satisfactory Instrumentation of I.C. Engines.** *Gas and Oil Power*, **64** (1968), p. 76 (Mar.), and p. 172 (Apr.) [3 pp., 5 phot.]

Part I of this articles deals with the requirements of instruments intended for use in the development, testing, or operation of i.c. engines, with particular reference to Diesels. This has been the subject of 20 years' research by the British Internal Combustion Engine Research Institute (BICERI), and has led to the development of a group of instruments for the monitoring and control of Diesels, both stationary and mobile.

Many difficult conditions have to be overcome by such instruments, notably vibration, high temperature, and rapid changes of speed and pressure; and the readings of any particular quantity have often to extend over a wide range of values. Experience has shown that most instrument failures on the transducer side of engine instruments are mechanical rather than electrical, and BICERI have concentrated on the development of " tough " transducers. To ease the user's task they have also planned their display units ergonomically to suit assembly in modules, or vertical stacks mounted in tiers.

Fault finding by audible means can be useful, and there is scope for the development of instruments working on this principle with better precision than the human ear. Testing for noise raises other problems, and it is recommended that every design team should include a noise engineer or be in constant touch with one. It is probably much cheaper to design *ab initio* for an adequately low noise level than to have to cure excessive noise afterwards.

Multiplicity of readings may tend to confuse operators, and many of these readings can be avoided over considerable periods by an instrumentation system which shows deviations from a standard or average reading for periodic checks or for data logging. In some BICERI equipment, therefore, data logging can be incorporated to show only any deviation

from an approved pattern. For example, with a multi-cylinder engine, it may be sufficient to record continuously from one cylinder only, and to scan the others at intervals.

In Part II, the first six classes of BICERI instruments available to users are described, with a number of illustrations. They are manufactured and distributed by Unit Automation Ltd, of Tonbridge, and are known as U.A.L.—BICERI instrument modules.

Module No. 1 is a pyrometer for use with chromel–alumel thermocouples of range 60° to 800°C or 60° to 1,000°C. It has a 3¼-in diameter dial indicator and a six-way switch. *In-situ* adjustment of line resistance can be carried out and the instrument voltage can be verified against external reference. The instrument can therefore be set up, ready for use on all six channels, in about 15 minutes. Without these *in-situ* adjustment provisions a much longer time would be needed.

Module No. 2 is similar to No. 1 but has a 20-way switch.

Module No. 3 is a twin-unit pyrometer arranged for the simultaneous display, with other similar modules, of a number of temperature measurements when switching is not acceptable. Otherwise the ranges, sizes, and adjustment arrangements are similar to those for Modules 1 and 2.

Module No. 4 is a six-channel or 20-channel transportable thermocouple pyrometer for use when the matching of thermocouples and their leads to given indicators is difficult or where coupling resistances exceed 25 ohms. The temperature ranges are 0°–800°C or 0°–200°C.

Module No. 5 embodies resistance thermometers with switching for six or 20 channels. Standard temperature ranges of the platinum resistance elements are 60°C to +60°C, 50°C to +100°C, 50°C to +150°C, 0°C to 100°C, 0°C to 150°C, 20°C to 120°C, 0°C to 200°C, 0°C to 250°C, and 0°C to 300°C.

Module No. 6 is a twin-unit instrument similar to No. 5 except that it has two indicators instead of one. It is intended for the simultaneous display, with a number of others of this type, of several temperature measurements by resistance bulbs.

DECK MACHINERY AND CARGO HANDLING

(*See also Abstracts No.* 26,798, 26,799, *and* 26,839)

26,830 **Transport of Liquefied Gases by Sea.** KLÄY, H. *J. Refrig.*, **11** (1968), p. 120 (May), and p. 141 (June) [4½ pp., 9 ref., 1 graph, 4 diag.]

Two types of liquefied gas are discussed in Part I of this paper, namely Liquefied Petroleum Gas (LPG), and Liquefied Natural Gas (LNG). Gases of type LPG, with which the Author is mainly concerned, are hydrocarbons such as ethane, propane, butane, propylene, and butadiene. They are all gases in normal atmospheric conditions but can be liquefied by pressure alone without any cooling. Ethylene, with a critical temperature of +9·5°C, almost falls within this group. For the purpose of marine transport, ammonia can also be included, although it is not a hydrocarbon. The commercial varieties of these gases (as transported) are not pure, and their thermodynamic properties may differ considerably from those of the pure gases. LPG is obtained mainly during the processing of crude oil.

LNG is a mixture of 80–90% methane with nitrogen, helium, and other

gases. It can be transported in its natural gaseous form through pipelines, or in liquefied form in tankers at atmospheric pressure and a temperature of 165 C. For this, it has to be liquefied before shipment. Its critical temperature is 82 C, and it can therefore not be liquefied by pressure alone. Its transport demands highly specialised tankers, which the Author does not discuss.

The great advantage of transporting petroleum gas in liquid form is the great saving of space aboard; for example, 1 cu ft of propane gas at 0 C and 14·7 lb sq in abs. occupies about 1 290 cu ft when liquefied, with a density of about 37 lb cu ft.

LPG tankers are of three types: pressure tankers with no refrigeration, semi-refrigerated pressure tankers, and fully-refrigerated tankers. In the second and third types the tanks are thermally insulated, but some heat will always be gained by the cargo. This must be removed by a refrigerating plant which can also be used for cooling the cargo if it is taken from land storage at a higher temperature than that used during transport. The principles and methods of operation are briefly described.

Pressure tankers without refrigeration are generally small vessels with a cargo capacity less than 1,000 cu m (35,300 cu ft), and the tanks are designed for a saturated-vapour pressure corresponding to a boiling point of 45 C. For butadiene this is about 5 atmospheres, for propane about 15 at., and for propane and ammonia about 21 at. If the gas is stored on land under the same conditions, the ships can be loaded and unloaded by pumps and neither thermal insulation nor refrigeration is needed. But usually, because of the high cost of thick-walled pressure containers, the gas is stored at a lower pressure and a refrigerating plant has to be installed on land. Loading is then done by pumps, and unloading can be done by the Sulzer labyrinth-piston compressor which is described in considerable detail in Part II of the paper. In addition to its use for this purpose, this compressor enables the vapours produced by heat gain during the voyage, and by throttling during unloading, to be drawn off and reliquefied without contamination by oil. (See also Abstract No. 15,446, June 1959.)

Semi-refrigerated pressure tankers are usually of medium size and of cargo capacity between 1,000 and 5,000 cu m. The pressures are chosen so that the corresponding temperatures do not need cargo tanks made of special low-temperature materials. If the service pressure for which the tanks are designed is such that butane can be transported without refrigeration, the cargo temperature for the other gases carried usually lies between 10 C and 10 C.

Fully-refrigerated tankers usually have a capacity in excess of 5,000 cu m. The pressure in their cargo tanks is between 0·05 and 0·3 kg sq cm (0·71 and 4·27 lb/sq in) above atmospheric, so that the walls can be much thinner than those of pressure tanks. But they must be made of low-temperature steels and well insulated thermally; the temperature for commercial liquid propane may be as low as 50 C.

Part II of the paper describes, in addition to the Sulzer labyrinth-piston compressor, the liquid-gas tankers *Arctic Propane*, *Lincoln Ellsworth*, and *Havgas*.

The *Arctic Propane* is a semi-refrigerated pressure tanker built by A/S Moss Værft & Dokk of Norway for Bulkship A S & Co. She has six cylindrical tanks (four below deck) taking in all 3,000 cu m of liquid

propane or ammonia. The cargo can be taken aboard from fully-refrigerated pressureless tanks on land at −50° C or from pressure tanks at −45° C. On board, the cargo, whose temperature is maintained at −10° C, can be heated from −50° C to −5° C or cooled from −45° C to −10° C. The reliquefaction plant comprises three single-stage labyrinth-piston compressors, each with a suction capacity of 270 cu m/hr. The ship is propelled by a six-cylinder Diesel of Sulzer type 6TAD36, developing 1,560 h.p. at 300 r.p.m. She has a service speed of 12·4 knots.

The semi-refrigerated *Lincoln Ellsworth* and the fully-refrigerated *Havgas* have been described at length in previous articles covered respectively by Abstracts No. 25,599 (Aug. 1967), and 24,601 (Sept. 1966).

VIBRATION AND SOUND-PROOFING

26,831 **Vibration of a Huge Tanker with Special Consideration of Athwartship Flexibility.** KUMAI, T. *Appl. Mech. Res. Inst. Kyushu Univ. Reports*, **15** (1967), p. 101 (No. 51) [8 pp., 2 ref., 1 tab., 5 graphs, 3 diag.]

This is a full translation into English of the paper covered by Abstract No. 25,847 (Oct. 1967).

26,832 **Vibration Isolation of Elastically-Mounted Marine Engines** (in German). HAHOLD, S. *M.T.Z.*, **28** (1967), p. 464 (Nov.) [6 pp., 18 ref., 4 tab., 3 graphs, 3 diag.]

As the usual type of marine-engine foundation is a framework, its mechanical impedance is less than that of the massive engine-foundations common in land practice; the two types of foundation therefore react differently to engine noise and vibration, and the difference is significant at frequencies above 100 c.p.s. A consequence of this is that the equations generally used in calculating the vibration- and noise-isolating effects of elastic engine-mountings are of only limited validity when applied to elastic mountings for marine engines.

The Author discusses the problem of establishing the value of the effective mechanical impedance, for use in marine engine-mounting calculations. Equations are derived for the approximate calculation of the effectiveness of the structure-borne noise isolation of ships' elastic mountings. Added masses fitted below the resilient elements are suggested as a method of increasing impedance and improving vibration-isolation. Data are given on the reduction in structure-borne noise to be expected from this method. The double-isolation method is also mentioned (see Abstract No. 26,456, May 1968).

CORROSION, FOULING, AND PREVENTION

(*See also Abstract No.* 26,847)

26,833 **Guidelines for Selection of Marine Materials.** TUTHILL, A. H., and SCHILLMOLLER, C. M. *Journal of Ocean Technology*, **2** (1967), p. 6 (Dec.) [31 pp., 23 ref., 17 tab., 6 graphs, 7 diag., 4 phot.]

The Authors present charts and summaries of published data and experience with metals and alloys in sea-water service, so as to permit the designer to make his initial selection of materials more easily. These charts and summaries cover general wastage, pitting, crevice effects, fouling, velocity effects and cavitation, galvanic effects, selective attack, stress-corrosion cracking, effects of deep immersion, and cost. The

behaviour of steel in the various near-surface and near-bottom zones (e.g. splash zone, mud line), and methods of protecting it, are briefly reviewed. Among the applications considered are sea-water heat-exchange systems, propellers, stern-tube bearings, deck fittings, fasteners, wire rope, buoys, and floating platforms.

26,834 **High-Temperature Corrosion Studies in an Oil-Fired Laboratory Combustor.** Barrett, R. E. *A.S.M.E., Paper No. 66-WA/CD-2, presented* 27 Nov.-1 Dec. 1966 [9 pp., 29 ref., 3 tab., 11 graphs, 4 diag.]

Interaction of sulphur oxides in the combustion gas with furnace components is known to be the major contributor to corrosion and deposits in boilers and gas turbines. These interactions are of two kinds: (*a*) Catalysis, whereby furnace components serve as catalytic surfaces to speed up the formation of SO_3 from SO_2 and molecular oxygen, and (*b*) chemical reaction, whereby the sulphur oxides react chemically with the furnace components or with deposits on them.

In the work here described, catalytic formation of SO_3 on bare and coated steel (carbon and alloy) test surfaces at superheater temperatures was studied in an intrinsically non-catalytic laboratory combustor. This combustor, which was fired with either natural gas or No. 2 fuel oil, simulated a boiler-furnace environment. The effects of metal composition, temperature, and various coating materials (e.g. Fe_2O_3, MgO, sulphates of sodium and magnesium, fly ash, kaolin) were examined. Some information was also obtained on chemical reactions of sulphur with bare and coated specimens.

The main conclusions are as follows:—

1. Ferritic alloys which oxidise to Fe_2O_3 are significantly catalytic despite the presence of alloying elements.
2. Coatings on metal surfaces can reduce catalysis, both by chemical reaction and by acting as a physical barrier to prevent combustion gas from reaching the metal surface.
3. Many coatings, including fly ash, are not significantly catalytic.
4. As coatings deteriorate with time and become more porous, increasing quantities of SO_2 reach the metal surface and are catalysed to form SO_3.
5. MgO coatings react readily with SO_3 to form $MgSO_4$.
6. Alkali trisulphates do not form in five hours on exposed surfaces at superheater temperatures in the presence of 40 p.p.m. of SO_3, even when Na_2SO_4, K_2SO_4, and Fe_2O_3 are present on the metal surface.

26,835 **Basic [Chemical] Problems in the Formation of [SO_3 and] Sulphates in Boiler Furnaces.** Reid, W. T. *A.S.M.E., Paper No. 66-WA/CD-1, presented* 27 Nov.-1 Dec. 1966 [4 pp., 7 ref., 2 tab., 5 graphs]

26,836 **Cathodic Protection by the Automatic Control of Impressed Current.** Ueda, K., Ogawa, H., and Yanagi, M. *Mitsubishi Heavy Industries Technical Review*, 4 (1967), p. 157 (No. 3) [7 pp., 4 ref., 2 tab., 5 graphs, 11 diag.]

The article gives general information on the theory of impressed-current cathodic protection, and includes some information on devices for the automatic control of the protective current. One of these is a newly-developed device which does not require a reference electrode.

OPERATION AND MAINTENANCE

(*See also Abstract No.* 26,781)

26,837 **The Current Revolution in Overseas Transportation.** ZUBALY, R. B., and LEWIS, E. V. *A.S.M.E., Paper No. 67 TRAN 33, presented* 28–30 Aug. 1967 [11 pp., 19 ref., 3 tab., 1 graph]

The paper discusses the recent change in the philosophy of shipping companies engaged in overseas transportation; emphasis is now laid on the overall movement of cargo from inland point of origin to ultimate destination rather than merely from pier to pier. An outline is given of some pioneer unitised-cargo operations (Seatrain, Sea-Land, Matson Line, Grace Line), of the beginnings of intensive container traffic on the North Atlantic route, and of plans for the Pacific. This is followed by a short review of some of the problems involved in unitisation (compatibility with land transport, ship operator's organisation, ship and terminal design, documentation). The economic significance of these new developments is then considered, with particular reference to time and cost savings to shippers, the resulting stimulation of world trade to be expected, and potential benefits to the economies of both developing and industrial countries. Data are given to show how reductions of a few per cent in overall distribution cost and/or delivery time could make many more U.S. products competitive when exported to Europe or elsewhere.

26,838 **Research Philosophy for Transportation [Application of " Systems Engineering "].** BAKER, R. F. *A.S.M.E., Paper No. 67-TRAN-47, presented* 28 30 Aug. 1967 [5 pp., 1 ref.]

26,839 **Mediterranean Container Clearance Centre.** BOURNE, G. B., EMERSON, E. C., and FOXWELL, C. *Containerisation International,* **2** (1968), p. 14 (June) [2½ pp., 2 tab., 4 diag.]

Trends in world transport suggest that transoceanic routes in the future may serve only one or two clearance centres in each trade zone. These centres will then serve industrial areas within each zone for trade to other parts of the world. It seems likely also that most of the traffic handled by each centre will be in full containers for transshipment, and will not require local distribution.

Against this background, the Authors examine the case for one or more container clearance centres in the Mediterranean, basing their arguments on the figures for 1965 (which they quote) of the values of trade between Mediterranean countries and other regions of the world. These figures indicate that in that year a Mediterranean container clearance centre would have supported the following services: –

Region	*Containers per year (one way)*	*Ships per week (one way)*
U.S. and Canada	235,000	2·35
South America	85,000	0·85
North East Europe	177,000	1·77
U.S.S.R.	189,000	1·89

These figures assume that the trade from each of the countries surrounding

the centre is carried from its major ports to the centre by feeder services, and that each ship carries 2,000 containers. For fully-balanced conditions, the total numbers of containers required would be twice those given in the table.

From these data the Authors estimate that, if all major ports in the Mediterranean used such a clearance centre, nearly 3 million containers per year would be handled by 1970; however, political, social, and other considerations will probably reduce this figure to about 1 million. The centre should be so located as to minimise total transport costs per container-mile, which will include the feeder-service cost as well as the trans-ocean cost. Other considerations affect the choice of site; for example, ports near the Atlantic access point may not use the centre for west-bound traffic because it will offer them no cost reduction. On the whole, the Authors estimate that two centres serving the Mediterranean ports, one at Marseilles and one at Crete, would offer a slight advantage in the total number of container-miles incurred per year over one centre, which would be best located in Sicily.

The advent of extensive container traffic will intensify the need for rapid turn-round of cargo at all stages. Container ports will take the nature of mechanised landing bays transferring cargo from ship to ship, or from ship to inland clearance depots for Customs examination and unpacking. Roll-on/roll-off container loading has some advantages for low-density short-sea routes, but is not suitable for large-scale container operation. One disadvantage is the much greater quayside marshalling area needed as compared with the vertical loading system.

The Authors list the basic requirements for the port transhipment area, one of which is high-density packing of containers for quick handling. The ideal layout of the port, which must allow for future developments to avoid obsolescence, can be built up in multiples of a unit port, the number of units required depending on the expected trade. In the early 1970s, a Mediterranean clearance centre may handle between 10,000 and 20,000 20-ft containers per week, which will need up to ten large ships and 40 to 50 feeder ships.

The unit port illustrated in a diagram has 1 kilometre of quay, which would allow one large and three small ships to berth simultaneously. Behind the large transoceanic berth, 400 m long, is the container parking area for exports, and behind the three 200-m berths for the smaller feeder ships is the import parking area. Between the berths and the parking areas is a two-way conveyor system. Four gantry portal cranes per unit length of 1 km will transfer containers between ship and shore, and four cantilever gantry cranes will serve the container park, in which containers are stacked five high. The park is roughly 90 m wide, which gives a parking density of 10 containers per metre of quay at 70% loading.

26,840 **New Engineering Techniques for Application to Deep-Water Mooring.** GAY, S. M. *A.S.M.E., Paper No. 66–PET-31, presented* 18–21 Sept. 1966 [12 pp., 16 ref., 1 tab., 8 graphs, 5 diag.]

This paper is mainly concerned with the theory and calculation of three-dimensional equilibrium configurations assumed by heavy anchor or towing cables in deep water under the influence of current, etc. A computer program is outlined and some typical results (for towing a paravane at depths to 1,000 ft) are presented.

26,841 **Investigation of Embedment Anchors for Deep-Ocean Use.** SMITH, J. E. *A.S.M.E., Paper No. 66-PET-32, presented* 18–21 Sept. 1966 [14 pp., 2 ref., 3 tab., 13 diag., 7 phot.]

Two commercial types of explosive-embedment anchors (which, on nearing the bottom, are fired into it by a gun-reactor unit) were tested by the U.S. Naval Civil Engineering Laboratory to examine their suitability for deep-ocean anchorage systems. The anchors are described and illustrated. Tests were conducted in both sand and mud bottoms, in shallow water and in depths to 6,000 ft. Components of the anchors were subjected to pressures up to 10,000 lb/sq in. in the laboratory. Successful discharges and embedments were achieved with one anchor type in depths to 6,000 ft, and with the other in depths to 1,200 ft. A large percentage of the tests resulted in malfunctions or in holding capacities below the nominal rating. However, malfunctions were traced to correctable causes in most cases, and the low holding powers were attributed partly to inadequate control of anchor attitude when discharged into the bottom and to other operational problems. It is concluded that the principle of explosive-embedment anchors offers reasonable potential for development of deep-ocean anchors that will be better than any now known to exist. Both commercial types of explosive anchors possess approximately equal potential, but have different deficiencies to overcome.

26,842 **Determination of the Optimum Scope [Length of Mooring Line/Depth of Water] of a Moored Buoy.** MORROW, W. B., and CHANG, W. F. *Journal of Ocean Technology*, **2** (1967), p. 37 (Dec.) [6 pp., 5 ref., 3 graphs, 3 diag.]

A method is derived for calculating optimum scope, the criterion being minimum cable tension for some pre-selected current-velocity distribution over the water depth.

26,843 **[Simple Formulae for] Choosing the Optimum Distance between an Icebreaker and a Following Vessel during Pilotage [through a Solid Ice Sheet]** (in Russian). BABAIN, V. *Morskoi Flot*, **28** (1968), p. 15 (Jan.) [1 p.]

The Author is concerned mainly with operations on the Northern Sea Route. The main factor governing the distance between the two vessels is the speed with which the ice closes in behind the icebreaker. This speed depends mainly on the speed and direction of the wind.

26,844 **Optimisation of the Major Mechanical Overhaul Interval from Reliability Studies.** BRAGAW, L. K. *Soc. N.A.M.E. (New England Section), paper read* May 1965 [44 pp., 25 ref., 17 tab., 8 graphs, 9 diag.]

The maintenance costs of a machinery system can be reduced and its reliability improved by "optimising" the intervals between major overhauls. This can be done by the methods of operations research. Various overhaul and maintenance policies are possible, depending largely on the accessibility of components. The Author discusses the application of reliability and maintenance theory to the development of component and group reliability functions for use in optimising the major overhaul cycle. He accepts for the definition of reliability "the probability of a device performing its purpose adequately for the period of time intended under operating conditions encountered", and points out that it is expressed as a dimensionless number between zero and unity.

The reliability function is defined as the limiting value of the ratio of number of successes to the total number of events as the latter tends to infinity.

The Author describes techniques for determining the reliability functions for individual mechanical components and for a component group, and then sets out to determine the optimum major overhaul policy for a mechanical system, based mainly on the optimum overhaul policy of the least accessible group of components. To illustrate the method, the Author has chosen the Cummins VT12M marine Diesel engine, of which 140 are installed in U.S. patrol boats; data for the analysis were obtained from a variety of sources, including service records. In classifying the data, failure was defined as " inability of a component or machine to perform its designed function or output ", and therefore included not only actual breakdown, but also excessive wear.

The Author then defines some of the terms used in Reliability Mathematics, referring the reader to more detailed treatments in published literature; these include discussions of Reliability of Series and Parallel Systems, Reliability Distributions Characteristic of Various Modes of Failure, Data Analysis Techniques, and Development of a Reliability Function for a Component and for a Mechanical System.

Dealing with the determination of Major Overhaul Policy and Scheduling, the Author dismisses the policy of performing no major overhaul, but applying corrective maintenance as each failure occurs; this has been shown to be inadvisable. The two other policies are (*a*) to perform major overhauls at specified intervals, irrespective of intervening failures, and (*b*) to perform major overhauls at a specified interval after each failure. The choice here depends largely on the accessibility of components. Components and subsystems that can be maintained without major dismantling are items for preventive maintenance rather than major overhaul. For the Cummins VT12M, the water pumps, fuel pump, lubrication pump, injectors, and tappets fall into this category. Other items can be exposed and given a " top overhaul " by removing the cylinder heads. But the piston, piston rings, cylinder liner, connecting rod, bearing shells, and crankshaft, which he terms the central group, can be inspected only after disassembly. He defines disassembly, inspection, and maintenance of this central group as a major overhaul. The reliability function of this group will determine whether policy (*a*) or (*b*) should be followed.

Reliability functions for each component of the central group of the Cummins engine are derived, the processes being described in some detail, and from them an overall Central Group Reliability Function is calculated. For this engine this reliability function for the central group approximated closely to that for the cylinder-liner wear, and the mean of the two (0·95) corresponds to an interval between major overhauls of 6,800 hours.

The paper ends with a series of recommendations for further investigation and for the application of the methods described to determine the optimum major overhaul intervals for marine engines already in service and for those in a new type of ship. Three appendices are entitled, respectively: Typical Material Failure Report, Summary of Data and Calculations (for the various components of the central group of the Cummins engine), and Load Schedule, Cyclic Endurance Test (for eight-hour test cycles during 600 hours of total engine running time).

26,845 **Work-Time Studies aboard Belgian Fishing Vessels (2).** HOVART, P., and CLEEREN, G. *Fishing News International*, **6** (1967), p. 36 (Oct.) [3 pp., 2 tab., 3 diag.]

In the first part of this article (see Abstract No. 25,859, Oct. 1967), the Authors (of the Belgian Fisheries Research Station, Ostend) gave general information on work studies whose object was the more efficient use of crews in Belgian fishing craft, particularly when shooting and hauling the nets and when handling the catch. This second part of the article gives detailed information on work studies carried out in a stern beam-trawler, and presents the results obtained. Comparison of these results with those from other fishing methods is to be made in a later article.

26,846 **Hull Scrubbing is Made Easy and Cheap by System Developed in Australia.** *Canadian Shipping and Marine Engineering News*, **39** (1968), p. 29 (July) [1 p., 1 phot.]

A new mechanical system of scrubbing ships' hulls has been devised by the Diving Company Pty Ltd, of North Narrabeen, New South Wales, in conjunction with Lucas Industrial Equipment, of Sydney.

The divers use 12-in diameter rotary brushes of nylon, soft stainless steel, or brass, driven hydraulically. The hydraulic pump is driven by a 6-h.p. motor and is designed to stall on overload without stalling the motor.

The advantages claimed for this system over compressed-air brush drive are that the hydraulic system is not limited to dockside operation due to the weight of the machinery, and that a drop in the r.p.m. of the brush does not cause a proportional drop in power output. The system is virtually maintenance-free, and operating costs are low.

It has been estimated that, using this system, three divers can finish in 12 hours a task that would need 20 divers using scrapers. Some indication is given of savings in the costs of ship operation arising from the use of the system.

26,847 **Underwater Maintenance.** *Propeller*, No. 34 (1968) [4 pp., 4 phot.]

This article discusses both underwater cleaning and underwater painting of the hull. The normal cleaning method, using frogmen with powered rotary brushes, is described and some drawbacks noted.

International Paints have developed a method of cleaning the submerged part of a ship's side and the turn of bilge (growth does not normally occur on the bottom) from the surface, using divers with brushes only for bulbous bows and for rudders and propellers. The advantages of this system over the normal method are that the vessel can be cleaned in tidal and/or polluted waters where a diver could not work.

The method employed is to use a 23-ft long cylindrical brush mounted on a small boat having a Diesel engine for propulsion and brush power. The brush shaft is telescopic and can deal with draughts up to 40 ft. The rotating brush is forced against the hull by its own suction. In one run from stem to stern of a ship, this " Brushboat " can cover an area of 600 ft by 20 ft in about 30 minutes. Allowing for the frogmen's operations, the largest ships can be cleaned within about six hours. This method is considerably cheaper than using divers with hand brushes.

The " Brushboat " is transportable, and the one already in use in the U.K. can be made available to all U.K. ports.

Scrubbing appears to have no adverse effect on anti-fouling coats. Hard-film anti-foulings which can be re-activated by scrubbing have been developed for use in conjunction with special maintenance contracts now offered by International Paints in conjunction with its subsidiary, the Underwater Maintenance Co. These contracts guarantee, for a fixed price, maintenance of ship speed within given limits over a specified period.

The underwater application of paint is still in the development stage, but some success has been achieved where surfaces have been cleaned by grit-blasting rather than rotary wire-brushing. So far, all work carried out in immersed conditions is slower and more expensive than similar work carried out above water.

In addition to underwater cleaning, the Underwater Maintenance Co. provide other specialist facilities for carrying out underwater repairs, etc.

FIRE DETECTION AND PREVENTION

26,848 **Shipboard Fire Defence Systems.** *Shipbuild. International*, **10** (1968), p. 22 (Apr.) [3 pp., 3 diag., 3 phot.]

This article consists mainly of fairly full descriptions of the shipboard fire-detection and fire-fighting equipment marketed by nine British and one Swedish firms. In a brief introductory discussion of general features of such equipment, attention is drawn to the increasing need for rapid detection and raising of the alarm of the outbreak of fires, owing to the growing use of automatic electrical control systems (which themselves increase the fire risk) and of unmanned machinery spaces.

Broadly, the equipment can be divided into four classes, with the objectives of (1) detection; (2) alarm raising; (3) automatic extinction; and (4) manual extinction.

S. Dixon & Son Limited, Leeds, supply monitors, distribution boxes, 2½-in fire valves, branch pipes, and nozzles. The monitors are made in three sizes—2½, 3, and 4 in—and are all capable of 360° horizontal movement and of being elevated.

L. M. Ericsson Telemateriel AB, Stockholm, make the Ericsson HAA100 fire-alarm system, based on the use of a thermal detector which, at a fixed temperature, usually 70° C (158° F), produces a fused metallic contact and so completes an alarm circuit. By use of a number of these detectors the position of the fire can be indicated.

Fire Appliance Industries (Dundee) Limited. This company makes all the types of portable fire extinguishers approved by the B.o.T. It also designs, manufactures, and installs CO_2 extinguishing systems to B.o.T. requirements.

Graviner (Colnbrook) Limited. This firm's continuous fire-detection system is for use in the scavenge ducts of Diesel engines, and provides early warning of a scavenge-duct fire and excessive blow-past. The company also produces a similar system for detecting fires in boiler-air casings. The temperature-sensitive detector can be installed as an independent unit or as part of a centralised temperature-monitoring system designed to give temperature indication and overheat warning throughout the main engine and its ancillary equipment. (See also Abstract No. 23,017, May, 1965.)

The Walter Kidde Company Limited, Northolt, make various types of

equipment, including smoke detectors (mainly for ships' holds); Marine-Zone type detection systems for accommodation and store-rooms; Kidde-Lucas inert-gas extinguishing systems for use in cargo spaces; total-flooding CO_2 systems for machinery spaces; and high-expansion foam equipment. Kidde Hi-Ex foam develops 1,000 volumes of foam for each volume of water, and therefore requires much less water for a given volume of foam than conventional foams with expansion ratios of about 8.

Mather & Platt Limited, Manchester. This firm's Grinnell Automatic Sprinkler and Fire Alarm system is designed for the protection of accommodation, public rooms, entrances, alleyways, and service spaces. The water sprinklers are automatically operated by the temperature rise caused by the fire, and at the same time an alarm bell is sounded.

The firm also make the Mulsi Spray system, using special water sprayers for fighting fires in boiler rooms and machinery spaces.

Minerva Detector Company Limited, Twickenham. Minerva fire-defence systems use a gas detector which reacts to the invisible combustion products formed in the early smouldering stages of a fire. It is particularly suitable for protecting machinery spaces, cargo holds, electrical equipment, and public rooms. The general protection it affords can be augmented by the use of air-sampling probes in the air-conditioning system. The firm also make a variety of fixed fire-extinguishing equipment, including automatic water sprinklers and manual and automatic CO_2 systems.

Nu-Swift International Limited, Yorkshire. The Twin-Thirty Dry Powder Unit model 1660 is an efficient extinguisher of inflammable-liquid fires which can occur in boiler rooms and machinery spaces. It can deal with two fires of spilt inflammable liquid covering 250 sq ft without being recharged.

J. & S. Sieger Limited, Poole, manufacture a number of gas-detection systems for revealing gas or vapour leaks in tankers carrying highly-volatile liquid cargoes or liquefied gas.

Merryweather & Sons Limited, London, supply two types of equipment. One type is for vessels such as fire-fighting tugs which are used basically to fight fires in other vessels. This equipment includes pumping sets and monitors of all types, including remote-controlled units. For fire protection within a vessel, Merryweather can engineer systems using foam or CO_2. These systems are designed to meet any special requirements of the particular vessel.

MISCELLANEOUS

26,849 **Experimental Measurement of Stresses while Laying Pipe Offshore.** Stewart, T. L., and Fraser, J. P. *A.S.M.E., Paper No. 66–PET–24, presented* 18–21 Sept. 1966 [8 pp., 1 tab., 6 graphs, 5 diag.]

The usual techniques of laying underwater pipelines from a specially-equipped " lay barge " are first outlined. In one the pipeline is supported and guided almost to the bottom by a device known as a " pontoon " or " stinger "; this is a long member sloping down from the stern of the barge and composed of hinged segments of adjustable buoyancy. The main purpose of this device is to avoid excessive bending. It is, however, also possible to lay pipe with no support other than longitudinal tension and a bend-limiting shoe aft of the barge.

An account is given of strain-gauge measurements to determine the

stresses in pipe during laying of offshore lines, in order to establish the adequacy of these two techniques. The measurements were made on four lines (ranging in diameter from 8 to 20 in) laid in the Gulf of Mexico in water depths up to 275 ft. Significant results are presented graphically and discussed. The techniques usually limited strain to less than the nominal yield of 0·5%, but permitted strains beyond the elastic limit. Wave action, even in calm seas, accounts for as much as 10% of total strain in the pipe.

26,850 **Pipe Stresses Induced in Laying Offshore Pipeline.** WILHOIT, J. C., and MERWIN, J. E. *A.S.M.E., Paper No. 66-PET-5, presented* 18-21 Sept. 1966 [7 pp., 2 ref., 5 tab., 4 diag.]

Underwater pipelines are commonly laid from a specially-equipped barge, most of the submerged string (which is weighted by concrete) being partly supported by a curved guide known as the " stinger ". The stinger extends obliquely downwards from the stern of the barge, and its configuration can be controlled by varying the buoyancy of its sections.

Part 1 of this paper considers the bending stresses induced in the unsupported length of concrete-coated pipe when it leaves the stinger with no change in sign of curvature. The influence of various parameters on the stresses is analysed. Part 2 considers the case which occurs when the pipeline tips over the end of the stinger and undergoes reversed curvature.

The results of computer calculations of geometry and stress for various conditions are tabulated.

26,851 **A Light-Media Pump for Dredging Minerals from the Sea Floor.** BALL, J. *A.S.M.E., Paper No. 67-UNT-4, presented* 30 Apr.-3 May 1967 [4 pp., 3 ref., 8 diag., 1 phot.]

The paper describes a new type of apparatus specifically designed to dredge sea-floor minerals (e.g. manganese nodules) from very great depths. This dredging system relies on the fact that the weight of a column (near-surface to near-bottom) composed of a mixture of nodules, sea water, and a light fluid such as kerosene can be made less than that of a column of sea water alone. A near-bottom unit scrapes nodules from the ocean floor, screens them, and passes them to a mixing chamber at the foot of the long vertical lift pipe. Kerosene or other fluid is forced down to the mixing chamber through a parallel pipe, by a high-pressure pump aboard the dredging barge. The near-bottom unit and the piping have built-in hydrostatic support. The lift pipe is not subject to any large pressure differential. The barge, and the entire apparatus, are towed behind a ship. The factors which determine the equipment size and the pump power required for varying mining rates and depths of water are discussed.

26,852 **[Measurements of] Task Accomplishment Times in Underwater Work.** STREIMER, I., TURNER, D. P., and VOLKMER, K. *Journal of Ocean Technology*, **2** (1968), p. 22 (Apr.) [5 pp., 16 ref., 3 tab., 1 diag.]

NOTES

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